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THE
ENGINEERING
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THE ENGINEERING EDUCATOR

A COMPREHENSIVE, PRACTICAL AND AUTHORITATIVE
GUIDE FOR ALL ENGAGED IN THE ENGINEERING INDUSTRY

Edited by

W. J. KEARTON, M.Eng., A.M.I.Mech.E., A.M.Inst.N.A.

ASSISTED BY A LARGE NUMBER OF SPECIALIST
CONTRIBUTORS

VOLUME II

FULLY ILLUSTRATED WITH MANY PHOTOGRAPHS
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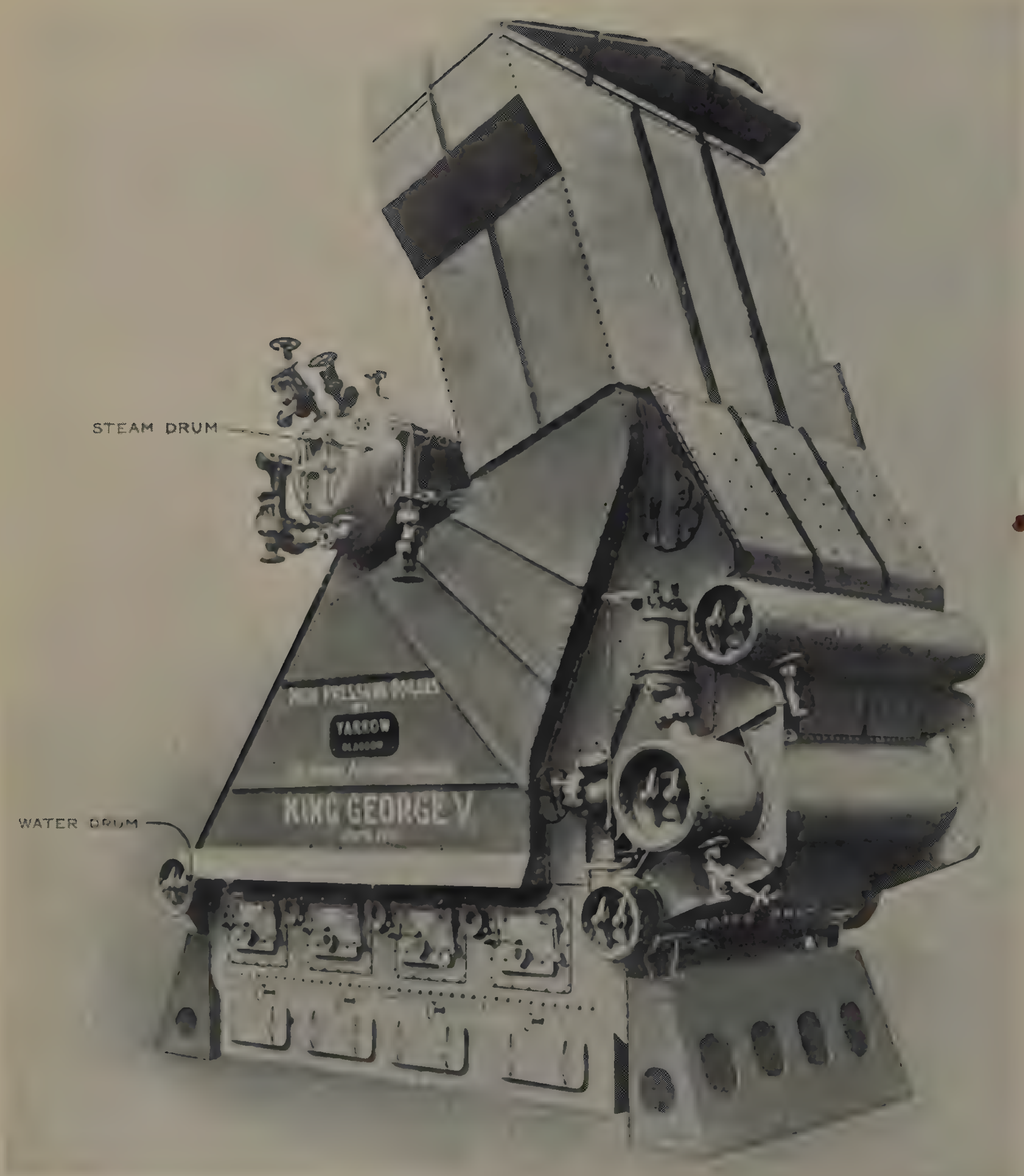

Fig. 25 High Pressure Yarrow Patent Water-tube Boiler with Superheater and Air-heater

Designed and Constructed by Yarrow & Co., Ltd., Glasgow
Test pressure 915 lb. per sq. in. Working pressure 575 lb. per sq. in.
Superheat 750° F.

(See the article on "Steam Boilers" which faces this page.)

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON IV

WATER-TUBE BOILERS—(*contd.*)

Vertical Water-tube Boilers. Vertical water-tube boilers may be divided into two main classes, viz., curved tubes and straight tubes.

In both cases, the general design usually consists of a number of tubes inclined at about 60° to the horizontal, and expanded direct into steam and water drums at the upper and lower ends respectively, the drums being set parallel to the front of the boiler. The number of drums used varies according to the construction of the plant, but in this country is usually not less than three or more than five, although advantage is sometimes taken to install an integral economizer, in which case six or even eight drums in all may be employed.

In the Ladd boiler, which has a certain popularity in America, only two drums are used, and these are set vertically one above the other, the central tubes being straight and the outer tubes curved at the ends. Similarly, the Kastner boiler, which has some vogue on the Continent, also relies on two drums only, the connecting tubes in this instance being all quite straight and rather longer than is customary.

In this country, however, the five- and three-drum Stirling boiler with curved tubes, the six- and four-drum Thompson boiler with straight tubes, and the three- and four-drum Yarrow boiler, also with straight tubes, are the best known of their respective types.

For connecting the tubes with the drums, simple expanded joints are practically universal practice. This form of joint depends for its permanent tightness on the two surfaces being perfectly clean at the time the tube is expanded, and also on the absence of burrs or cutting edges in the tube hole. The tubes are arranged to project inside the drums by about half to three-quarters of an inch, and are expanded by an expanding mandrel operated from the inside. Provided that the foregoing items are properly attended to, the actual expanding is a simple matter, and results in the formation of a thoroughly pressure-tight joint.

The design of vertical water-tube boilers does not lend itself to sectional construction, so that headers and tube caps are not employed. This latter consideration is sometimes cited as an advantage; it being contended that a multitude of caps is an objection, and that with the vertical construction, by opening one manhole, all tubes are immediately "get-at-able." On the other hand, the elimination of caps means that instead of all cleaning and renewal of tubes being an external operation this has to be done from inside the boiler drums. Apart, therefore, from the inconvenience of working in confined spaces, the question of time taken in cooling down the plant sufficiently to enable it to be worked in has also to be considered, when comparing relative advantages.

The Stirling Boiler. There are now two principal designs of the Stirling boiler, the standard five-drum plants (Fig. 20) consisting of three upper steam drums and two lower water drums, while the latter and more recent type as put forward for large-sized units, capable of evaporating up to 200,000 lb. of water per hour, consists of two upper steam drums and one lower water drum, as shown in Fig. 21.

The special feature of the Stirling boiler is the use of curved water tubes, the primary functions of which are to (1) provide a reliable means of expansion, (2) enable the true cylindrical form of the drums to be retained, (3) allow the tube ends to enter the drum plates at right angles and so ensure a perfectly expanded joint.

Access is obtained to each nest of tubes for internal cleaning or renewals by entering the steam drums, and a special chain-operated cleaner is employed for use in cleaning the curved tubes.

The upper drums are supported on a steel framing which is entirely independent of the brick setting. The tube nests and lower drums are all supported from the steam drums, so that the complete unit is suspended and free to expand in any direction.

All the tubes are of solid drawn steel, and are tested to a minimum pressure of 1,000 lb. per sq. in. for normal working conditions, the standard external diameter being $3\frac{1}{4}$ in. The tubes are arranged in parallel rows, with spaces between each pair of rows of sufficient width to allow of the removal and renewal of any tube without disturbing any other tube.

The tubes are expanded into their respective drum plates at either end, the tube ends being suitably bell-mouthed.

The system of baffling the furnace gases will be apparent from Fig. 20, each separate bank of tubes being passed in turn before the gases leave the boiler. The bulk of the evaporation takes place in the tubes nearest to the furnace, and the cold feed passes down the rear tubes into the lower water drum.

In the "tri-drum" design, the rear steam and the lower water drum are divided by patent diaphragms into compartments, in order to compel the incoming feed water to traverse two passes in the rear bank of tubes. These diaphragms are hinged to facilitate tube cleaning.

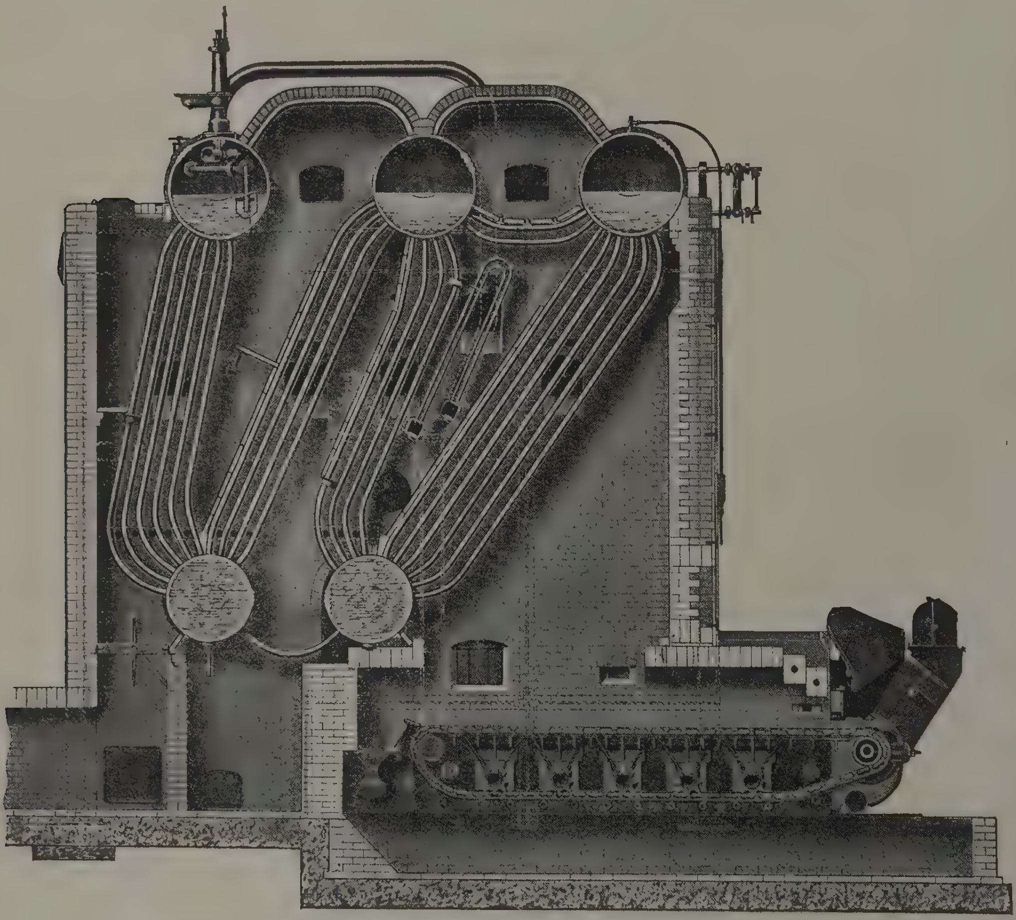

FIG. 20. FIVE-DRUM STIRLING BOILER

The Thompson Boiler. The standard "John Thompson" water-tube boiler has four drums, two upper and two lower, each upper and lower pair being connected together by banks of perfectly straight tubes, the two upper and two lower drums being connected horizontally by short curved connecting tubes.

To enable the straight tubes to enter the drum plates at right angles, the drums are pressed into a series of impressions where the tubes enter, the cylindrical shape of the drums being retained between the impressions, in order to maintain the strength of the drums. The inclination of the tubes nearest to the furnace is approximately 60° to the horizontal, while the second bank is somewhat steeper.

The two banks of tubes are baffled to enable the furnace gases to pass over them to the best advantage.

All the tubes are solid drawn and are 3¼ in. in diameter, and the complete boiler is suspended from structural steelwork to allow for expansion in any direction.

In the illustration given in Fig. 22, the boiler is shown with a superimposed economizer and integral superheater. and also a " Thompson " chain grate stoker.

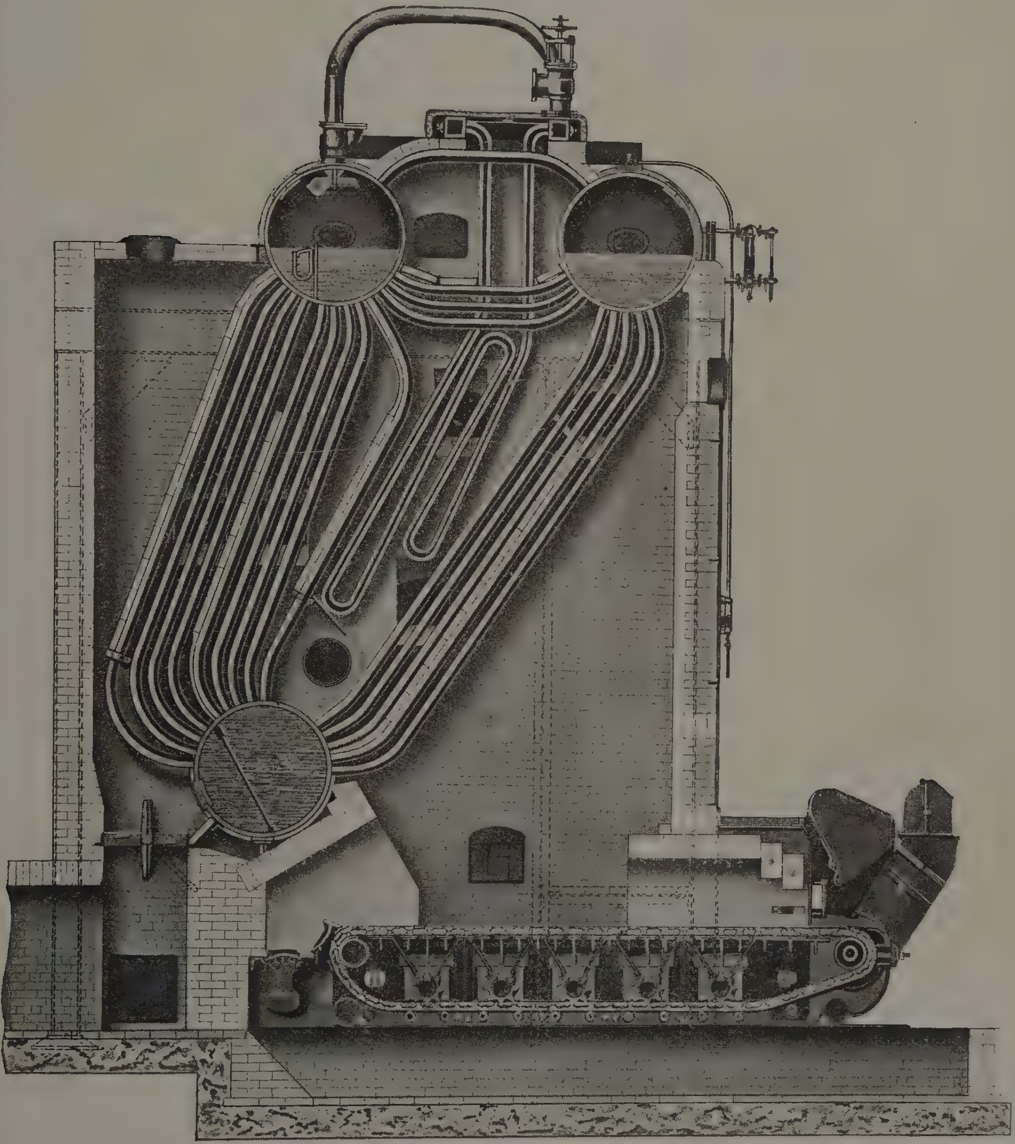

FIG. 21. STIRLING " TRI-DRUM " HIGH-DUTY BOILER

The Yarrow Boiler. The Yarrow boiler, although primarily designed for marine purposes, and particularly for use in T.B.D's. and similar craft, has been adapted for land conditions and installed at a number of plants.

The original design allowed for one upper steam drum and two lower water drums, each

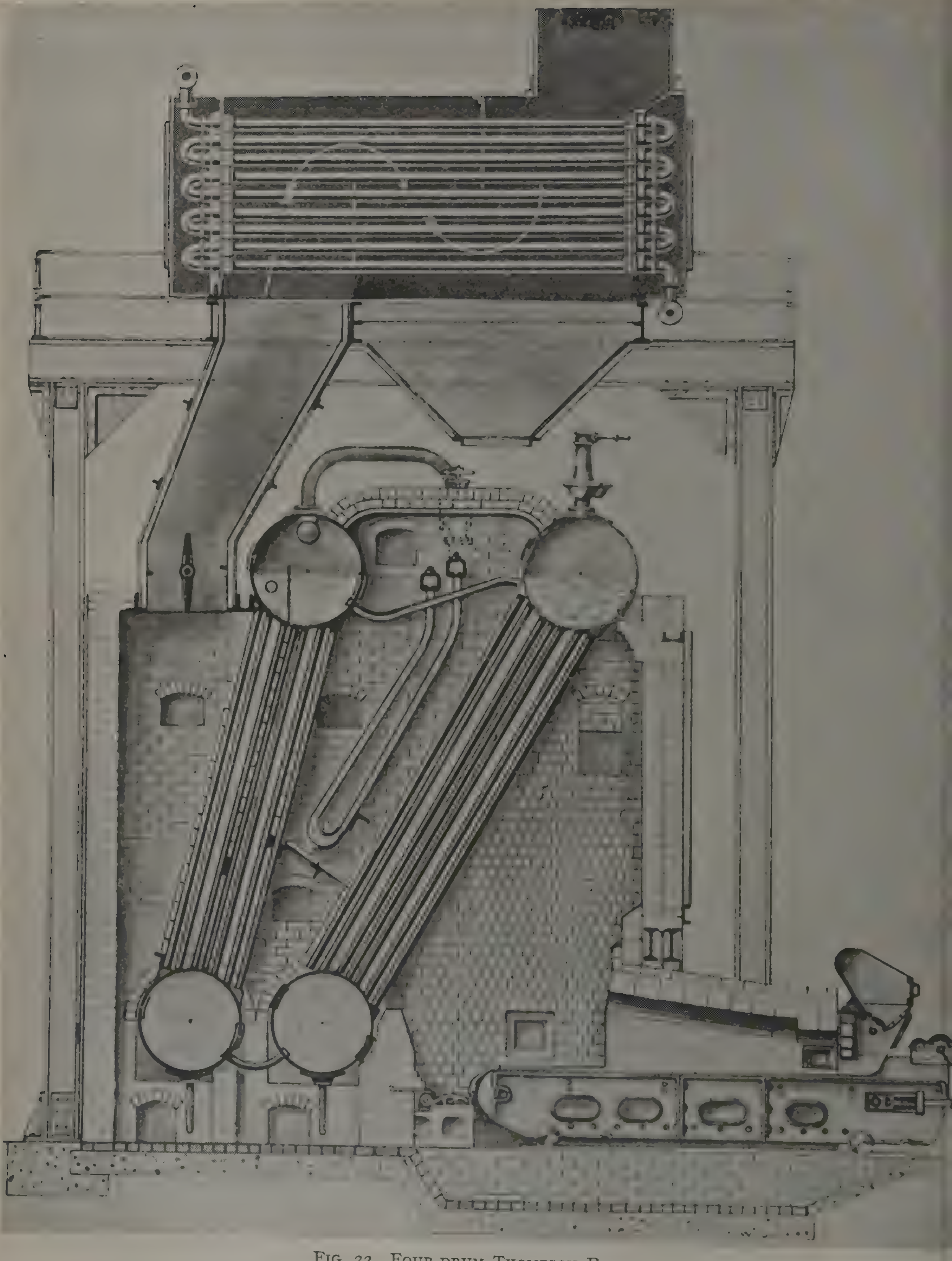

FIG. 22. FOUR-DRUM THOMPSON BOILER

of the latter being connected to the former by banks of straight small diameter steel tubes, the whole structure forming an inverted " V " with the furnace placed between the two banks of tubes.

For the land design the boiler has been slightly modified by placing the furnace at right angles to the drums, lifting one of them so that the furnace openings can be arranged beneath it, and introducing a third water drum at the rear of the second.

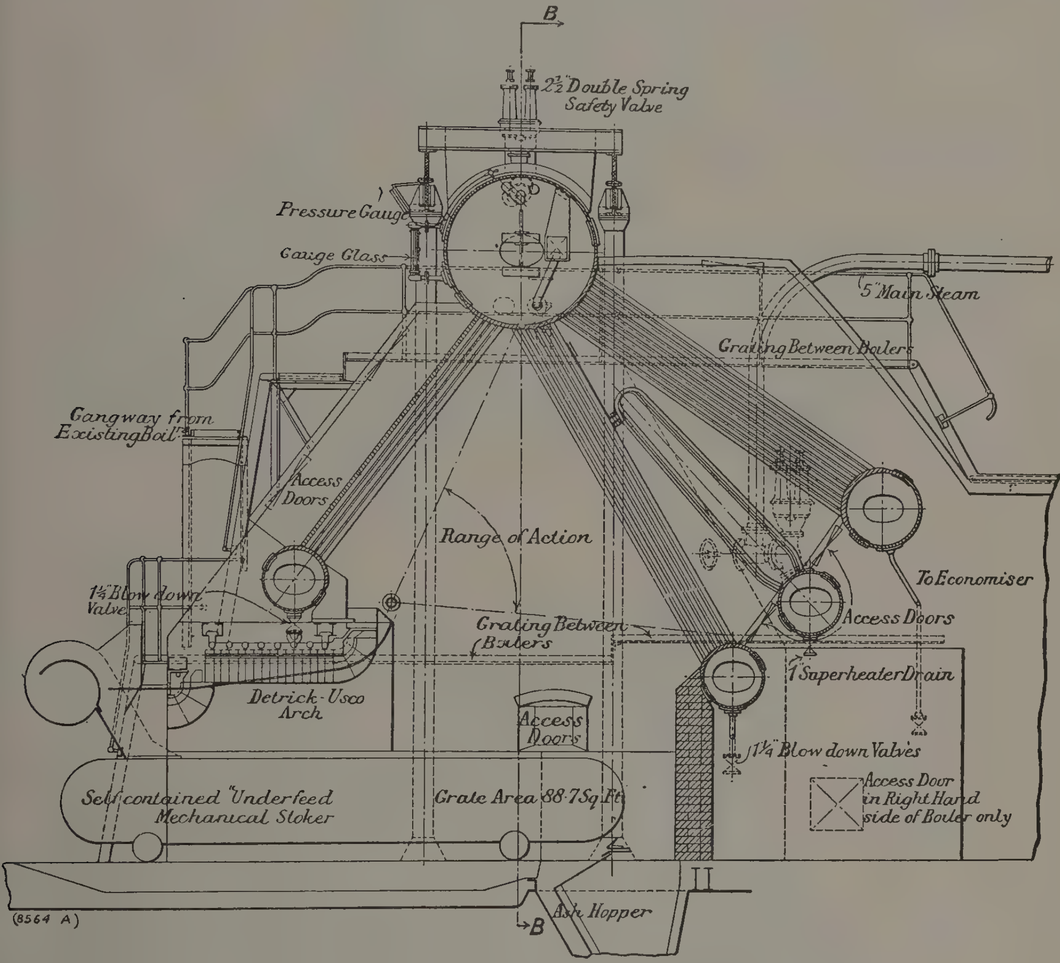

FIG. 23. CROSS-SECTION YARROW LAND TYPE BOILER

The special features of this boiler are the use of 1¾ in. diameter tubes, the entire absence of " baffling "—allowing a straight through pass for the gases—and a high rate of evaporation per square foot of heating surface.

It has been manufactured for evaporations up to 100,000 lb. of water per hour, and is particularly adaptable to the use of oil fuel.

In connection with the further development of the Yarrow boiler for marine purposes, at high working pressures, some particularly interesting modifications have recently been introduced.

The general arrangement of the boiler remains much the same as that adopted for land type practice, and the unit consists of an upper steam drum connected by straight tubes to three lower water drums, space for the superheater being provided between two of the latter.

To reduce riveted and bolted units to a

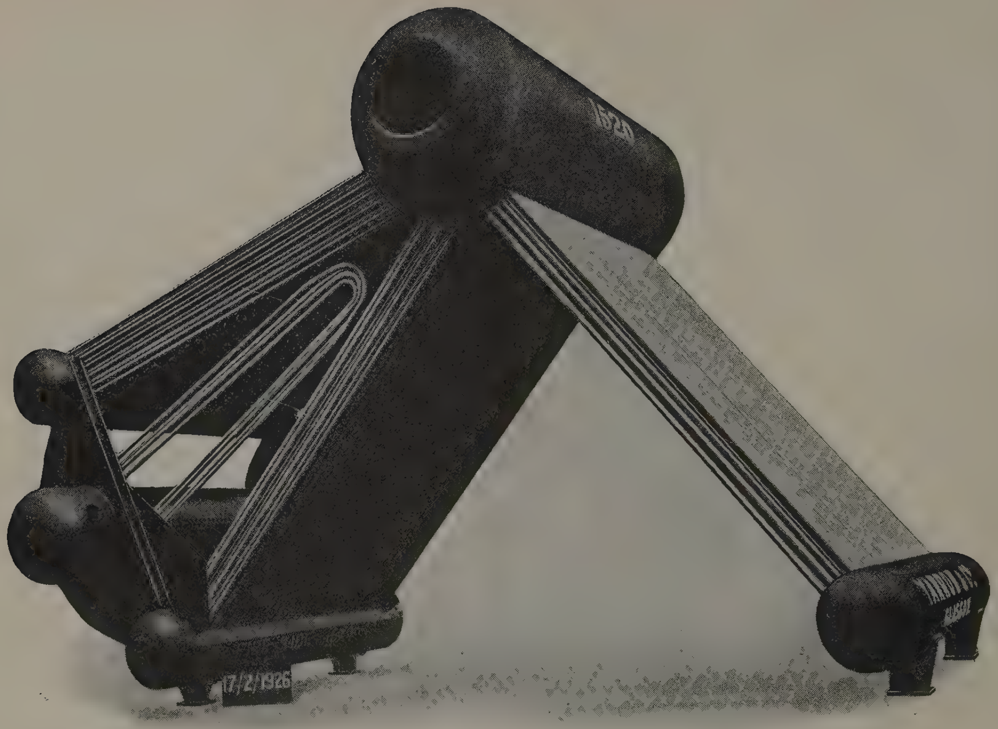

FIG. 24A

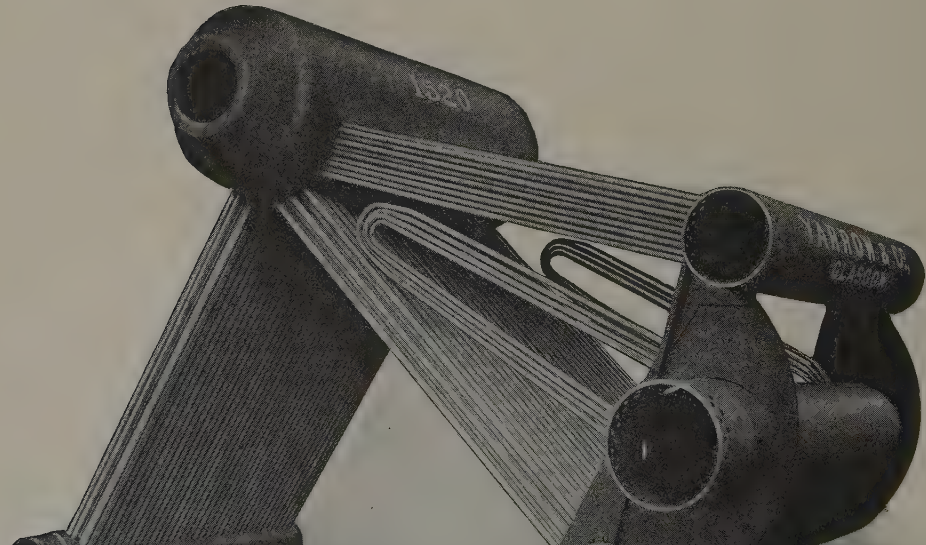

FIG. 24B. HIGH PRESSURE YARROW BOILER
Working Pressure 575 lb. per sq. inch Test Pressure 915 lb. per sq. inch

minimum, however, the steam drum is forged solid from a single ingot, and a description of this process is of interest.

A large octagonal ingot is forged in a hydraulic press into a round billet.

The billet, after careful examination for surface defects, is bored from end to end to take a mandrel.

The billet is then heated and forged into a ring, with an internal diameter slightly less than the finished dimension, while the length of the billet is not increased.

In the third forging operation a large mandrel is used, approximating in diameter to that of the finished drum, and the shell is forged to its required length without the diameter being increased.

The ends are left long and thicker than the main shell, so that after the interior has been machined to the desired diameter both ends can be finally forged down on to a mandrel of a little less diameter than the finished manholes. After being thoroughly annealed the drum is subjected to mechanical tests and finally machined.

The boiler in question, of which two have now been constructed for the Mercantile Marine, is operating at a working pressure of 575 lb. per square inch and has been subjected to a test pressure of 915 lb. per square inch.

APPLIED MECHANICS

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LESSON XI

STRENGTH OF BEAMS

In our previous work we have considered the expression $\frac{f}{y}=\frac{M}{I}=\frac{E}{R}$, together with the appropriate expression for the I of various sections, and we can now show the applications in a few examples on the strength of beams.

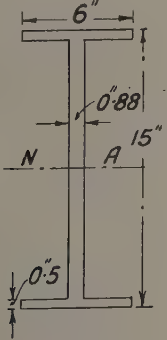

Fig. 53

Example. An I section beam, 15 in. × 6 in., flange 0·5 in. thick, web 0·88 in. thick, is supported over a span of 30 ft. Determine the maximum uniformly distributed load this beam will support if the tensile and compressive stresses are not to exceed $7\frac{1}{2}$ tons per square inch.

$I_{NA} = \frac{1}{3}\Sigma bd^3$

$I_{NA} = \frac{1}{3}[6 \times (7{\cdot}5)^3 - 5{\cdot}12 \times (7)^3] \times 2$

$I_{NA} = 517\frac{1}{3}$ in.^{4} units.

For a beam supported at the ends and uniformly loaded,

$$\text{B.M.} = \frac{Wl}{8} \qquad . \qquad . \quad (1)$$

where W = total distributed load.

Also;

$$\text{B.M.} = \frac{f \times I}{y} \qquad . \qquad . \quad (2)$$

Substituting (2) in (1) we obtain

$$\frac{W \times 30 \times 12}{8} = \frac{7{\cdot}5 \times 517\frac{1}{3}}{7{\cdot}5} \quad \therefore W = \underline{11{\cdot}5 \text{ tons}}$$

Example. A built-up beam section is formed of a channel, an I beam and a flat plate as shown in Fig. 54. The beam, loaded on the top, spans 25 ft.; if the load carried by the beam = 1·5 tons per foot run, calculate the maximum tensile and compressive stresses.

To determine the position of the neutral axis NA.

Whole area $= (11 \times \frac{1}{2}) + (30\frac{1}{2} \times \frac{3}{4}) + 12$

$= 40\frac{3}{8}$ sq. in.

Taking moments of areas about BB, we get,

$40\frac{3}{8} \times z = (12 \times 1 \times \frac{1}{2}) + (6 \times \frac{3}{4} \times 1\frac{3}{8} + (18\frac{1}{2} \times \frac{3}{4} \times 11) + (6 \times \frac{3}{4} \times 20\frac{5}{8}) + (5 \times \frac{1}{2} \times 21\frac{1}{4}) + (3 \times 1 \times 22\frac{1}{2})$

$\therefore\ z = 9{\cdot}37$ in. nearly.

We can now fix the position of the neutral axis and complete the dimensions on Fig. 54.

Moment of Inertia of Section

$I_{NA} = \frac{1}{3}\Sigma bd^3.$

$I_{NA} = \frac{1}{3}[12 \times (9{\cdot}37^3) - 5\frac{1}{4} \times (7{\cdot}62)^3 - 6 \times (8{\cdot}37)^3 + 5 \times (12{\cdot}13)^3 + 1 \times (14{\cdot}63)^3 - 5\frac{1}{4} \times (10{\cdot}88)^3].$

$I_{NA} = 3{,}109$ (in.)^4 units.

To determine f_c we must use $y_c = 14{\cdot}63$, and to determine f_t we must use $y_t = 9{\cdot}37$ in the equation.

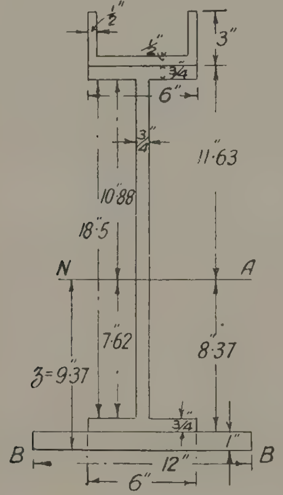

FIG. 54

$$\text{B.M.} = \frac{Wl}{8} = f \times \frac{I}{y}$$

$$\therefore\ \frac{1{\cdot}5 \times 25 \times 25 \times 12}{8} = f_c \times \frac{3109}{14{\cdot}63}$$

or $f_c = \underline{6{\cdot}618}$ tons per square inch.

$$\text{and again, } \frac{1{\cdot}5 \times 25 \times 25 \times 12}{8} = f_t \times \frac{3109}{9{\cdot}37}$$

$\therefore\ f_t = \underline{4{\cdot}242 \text{ tons per square inch.}}$

CONTINUOUS BEAMS

CLAPEYRON'S THEOREM OF THREE MOMENTS

A beam which is supported at more than two points is termed a *Continuous Beam.* The beam being simply supported at the two outer supports is free from bending moment over these supports, but the sections over the intermediate supports are subjected to bending moments. Clapeyron's equation enables us to calculate these moments.

Uniformly Distributed Load

Span ab; *Section PP* (Fig. 55)

Section PP is subjected to two bending moments: (1) that due to the moments over supports a and b, and (2) that due to the distributed load w_1, considering the span as being simply supported.

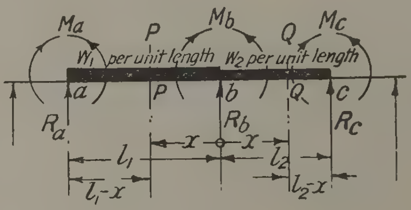

FIG. 55

We should here note that the bending moments on section PP, due to (1) and (2), are of opposite signs; that due to (1), the bending moment of the supports, tends to bend the beam to this shape: ⌢ and is here treated as +, whilst the bending moment due to the distributed load tends to bend the beam into this shape: ⌣ and is, therefore, treated as —.

We may now write—

Resultant B.M. on section PP

$$= \left[M_b + \frac{x}{l_1}(M_a - M_b)\right] - \left[\frac{w_1 l_1}{2}(l_1 - x) - \frac{w_1}{2}(l_1 - x)^2)\right]$$

Taking the origin over the support b, and considering x, measured to the left, as positive we can write—

$$EI\frac{d^2y}{dx^2} = M_b + \frac{x}{l_1}(M_a - M_b) - \frac{w}{2}(l_1 x - x^2) \quad (1)$$

$$\therefore\ EI\frac{dy}{dx} = M_b x + \frac{x^2}{2l_1}(M_a - M_b) - \frac{w_1}{2}\left(l_1\frac{x^2}{2} - \frac{x^3}{3}\right) + C \quad . \quad . \quad . \quad (2)$$

and when $x = 0$, $C = EI\dfrac{dy}{dx}$ at support b.

Again integrating, we get—

$$EIy = M_b\frac{x^2}{2} + \frac{x^3}{6l_1}(M_a - M_b) - \frac{w_1}{2}\left(\frac{l_1x^3}{6} - \frac{x^4}{12}\right) + Cx + D \quad . \quad (3)$$

and if we suppose that the supports are at the same level, then, when $x = o$, $y = o$, $\therefore D = O$.

Also when $x = l_1$, $y = o$, and substituting these values in (3) we get—

$$O = \frac{M_b l_1^2}{2} + \frac{l_1^3}{6l_1}(M_a - M_b) - \frac{w_1}{2}\left(\frac{l_1^4}{6} - \frac{l_1^4}{12}\right) + Cl_1$$

and substituting $EI\frac{dy}{dx}$ at b for C and dividing by l_1 we get—

$$EI\frac{dy}{dx} \text{ at } b = \frac{w_1 l_1^3}{24} - \frac{l_1}{6}(M_a - M_b) - \frac{M_b l_1}{2} \quad . \quad (4)$$

Span bc ; Section QQ

Similarly we may consider section QQ in span bc.

The resultant B.M. on section QQ = B.M., due to fixing moments M_b and M_c together with the B.M. due to the distributed load w_2, considering the span as simply supported.

Again, we take b as our origin, but x is positive when measured to the right.

$$\therefore -EI\frac{d^2y}{dx^2} = M_b + \frac{x}{l_2}(M_c - M_b) - \left[\frac{w_2 l_2}{2}(l_2 - x) - \frac{w_2}{2}(l_2 - x)^2\right]$$

$$\text{or} \quad -EI\frac{d^2y}{dx^2} = M_b + \frac{x}{l_2}(M_c - M_b) - \frac{w_2}{2}(l_2 x - x^2) \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$\therefore -EI\frac{dy}{dx} = M_b x + \frac{x^2}{2l_2}(M_c - M_b) - \frac{w_2}{2}\left(\frac{l_2 x^2}{2} - \frac{x^3}{3}\right) + A \quad . \quad . \quad . \quad (6)$$

and when $x = o$, $A = -EI\frac{dy}{dx}$ at b.

Integrating again we get—

$$-EIy = M_b\frac{x^2}{2} + \frac{x^3}{6l_2}(M_c - M_b) - \frac{w_2}{2}\left(\frac{l_2x^3}{6} - \frac{x^4}{12}\right) + Ax + B \quad . \quad . \quad (7)$$

When $x = O$, $y = O$. $\therefore B = O$

and again, when $x = l_2$, $y = o$ and substituting these values in (7)

$$\therefore O = \frac{M_b l_2^2}{2} + \frac{l_2^3}{6l_2}(M_c - M_b) - \frac{w_2}{2}\left(\frac{l_2^4}{6} - \frac{l_2^4}{12}\right) + Al_2.$$

$$O = \frac{M_b l_2^2}{2} + l_2^2\frac{(M_c - M_b)}{6} - \frac{w_2 l_2^4}{24} + Al_2$$

$$\text{or} - EI\frac{dy}{dx} \text{ at } b = \frac{w_2 l_2^4}{l_2 \times 24} - \frac{l_2^2(M_c - M_b)}{6l_2} - \frac{M_b l_2^2}{2l_2}$$

$$-EI\frac{dy}{dx} \text{ at } b = \frac{w_2 l_2^3}{24} - \frac{l_2(M_c - M_b)}{6} - \frac{M_b l_2}{2} \quad . \quad (8)$$

By addition of (4) and (8) we get—

$$O = \frac{w_1 l_1^3}{24} - \frac{l_1}{6}(M_a - M_b) - \frac{M_b l_1}{2} + \frac{w_2 l_2^3}{24} - \frac{l_2(M_c - M_b)}{6} - \frac{M_b l_2}{2}$$

which reduces to—

$$l_1(M_a + 2M_b) + l_2(2M_b + M_c) = \tfrac{1}{4}(w_1 l_1^3 + w_2 l_2^3) \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

which is Clapeyron's equation connecting the three moments of a continuous beam supporting uniformly distributed loads.

Example. A beam is continuous over three spans of 20, 40, and 30 ft., and carries uniformly distributed loads of 1, $1\frac{1}{2}$, and $1\frac{1}{4}$ tons per foot run respectively over the three spans as shown in Fig. 56. Determine the moments over the four supports and draw the B.M. and S.F. diagrams.

Applying Clapeyron's equation to spans ab and bc, we have—

$$l_1(M_a + 2M_b) + l_2(2M_b + M_c) = \tfrac{1}{4}(w_1 l_1^3 + w_2 l_2^3)$$

$$20(O + 2M_b) + 40(2M_b + M_c) = \tfrac{1}{4}[1 \times (20)^3 + 1\tfrac{1}{2} \times (40)^3]$$

$$\therefore 120\,M_b + 40M_c = 26{,}000 \quad . \quad . \quad . \quad (1)$$

Again, applying the equation to spans bc and cd we have—

$$l_2(M_b + 2M_c) + l_3(2M_c + M_d) = \tfrac{1}{4}(w_2 l_2^3 + w_3 l_3^3)$$

$$40(M_b + 2M_c) + 30(2M_c + O) = \tfrac{1}{4}[1\tfrac{1}{2} \times (40)^3 + 1\tfrac{1}{4} \times (30)^3]$$

$$\therefore \quad 40M_b + 140M_c = 32{,}437{\cdot}5 \qquad . \qquad . \qquad (2)$$

Combining (1) and (2),

$$\left.\begin{aligned} 120\,M_b + 40\,M_c &= 26{,}000 \\ 120\,M_b + 420\,M_c &= 97{,}312{\cdot}5 \end{aligned}\right\}$$

$$\therefore \quad M_b = \underline{154{\cdot}1} \text{ tons-ft., and } M_c = \underline{187{\cdot}7} \text{ tons-ft.}$$

Reactions

Taking moments of forces on span *cd* about support *c*,

$$M_c + (R \times 30) = 1\tfrac{1}{4} \times 30 \times 15$$

$$\therefore \quad R_d = \underline{12{\cdot}49} \text{ tons.}$$

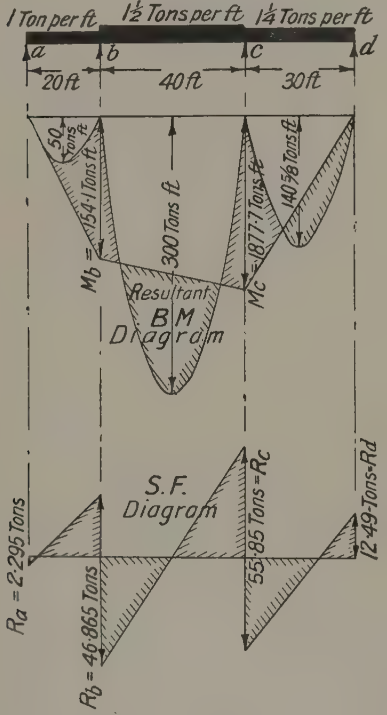

FIG. 56

Taking moments of forces on spans *bc* and *cd* about support *b*,

$$M_b + R_d \times 70 + R_c \times 40 = (40 \times 1\tfrac{1}{2} \times 20) + (30 \times 1\tfrac{1}{4} \times 55)$$

$$\therefore \quad R_c \quad = \underline{55{\cdot}85} \text{ tons.}$$

Taking moments of forces on span *ab* about support *b*,

$$R_a \times 20 + M_b = 20 \times \frac{20}{2}; \; \therefore \; R_a = \underline{2{\cdot}295} \text{ tons.}$$

Taking moments of forces on spans *ab* and *bc* about support *c*,

$$R_a \times 60 + R_b \times 40 + M_c = (40 \times 1\tfrac{1}{2} \times 20) + (20 \times 50)$$

$$\therefore \quad R_b = \underline{46{\cdot}865} \text{ tons.}$$

The resultant B.M. diagram is obtained by superimposing the diagram of bending moments due to the distributed loads on that due to the bending moments over the supports. Since these are bending moment diagrams of *opposite signs*, it is clear, that, where the diagrams cover each other, they cancel out, leaving the resultant B.M. diagram as shown etched in Fig. 56. The S.F. diagram should not present any difficulty, the reactions at the supports having been determined.

CONTINUOUS BEAMS. CONCENTRATED LOADS

When the loads are concentrated, the equation of three moments is written—

$$l_1(M_a + 2M_b) + l_2(2M_b + M_c) = -6\left(\frac{A_1x_1}{l_1} + \frac{A_2x_2}{l_2}\right)$$

where A = the area of the " simply supported " B.M. diagrams, and x = the distance of the centroid of these diagrams from the outer supports of the pair of spans under consideration. Now A_1 and A_2 are usually negative, being due to bending moments of an opposite sign to M_a, M_b or M_c; hence we write the above equation as—

$$l_1(M_a + 2M_b) + l_2(2M_b + M_c) = 6\left(\frac{A_1x_1}{l_1} + \frac{A_2x_2}{l_2}\right)$$

The following example is fully worked out in order to make the method of application clear.

EXAMPLE. A beam is continuous over three spans of 15, 21, and 12 ft. respectively, and carries loads of 4, 2, and 2 tons in the positions shown in Fig. 57. Determine the bending moment over each support, and the reactions at the supports. Draw the complete B.M. and S.F. diagrams.

Method. We first consider the spans as being discontinuous, and draw the appropriate B.M. diagrams. These are of the usual triangular form, as shown in Fig. 57; the maximum ordinates being $\frac{48}{5}$ tons-ft., $\frac{72}{7}$ tons-ft. and $\frac{9}{2}$ tons-ft. respectively.

We can now calculate A_1, A_2, and A_3 as follows—

(Units are tons and feet throughout)

Area of B.M. diagram for the left-hand span

$$= \frac{48}{5} \times \frac{15}{2} = A_1$$

Area of B.M. diagram for the centre span

$$= \frac{72}{7} \times \frac{21}{2} = A_2$$

Area of B.M. diagram for the right-hand span

$$= \frac{9}{2} \times \frac{12}{2} = A_3.$$

The distance x_1 of the centroid of the left-hand B.M. diagram from AA (obtained by taking moments of areas about AA) = 9 ft. = x_1; similarly we obtain $x_2 = 11$ ft.; $x_2{}^1 = 10$ ft. and $x_3 = 5$ ft. (see Fig. 57).

We can now apply Clapeyron's equation to the first pair of spans, viz., AB and BC, and we can write—

$$15(M_a + 2M_b) + 21(2M_b + M_c)$$
$$= 6\left(\frac{A_1 x_1}{l_1} + \frac{A_2 x_2}{l_2}\right)$$

On substituting the known numerical values in the above equation we obtain—

$$24M_b + 7M_c = 199{\cdot}54 \qquad . \qquad . \qquad (1)$$

Applying the equation to the second pair of spans, viz., BC and CD, we obtain—

$$21(M_b + 2M_c) + 12\,(2M_c + M_d)$$
$$= 6\left(\frac{A_2 x_2{}^1}{l_2} + \frac{A_3 x_3}{l_3}\right)$$

and substitution gives—

$$21M_b + 66M_c = 376{\cdot}07 \qquad . \qquad . \qquad (2)$$

Solutions of (1) and (2) determines the values of

$M_b = \underline{7{\cdot}333}$ tons-ft. and $M_c = \underline{3{\cdot}365}$ tons-ft.

Reactions

These are calculated by taking moments of forces about each support in turn, as follows—

Moments of forces about B
(left-hand side)

$$(R_a \times 15) + M_b = (4 \times 3)$$

$$\therefore \quad R_a = \frac{12 - 7{\cdot}333}{15} = \underline{0{\cdot}311} \text{ tons.}$$

Moments of forces about C
(right-hand side)

$$(R_d \times 12) + M_c = (2 \times 9)$$

$$\therefore \quad R_d = \frac{18 - 3{\cdot}365}{12} = \underline{1{\cdot}219} \text{ tons.}$$

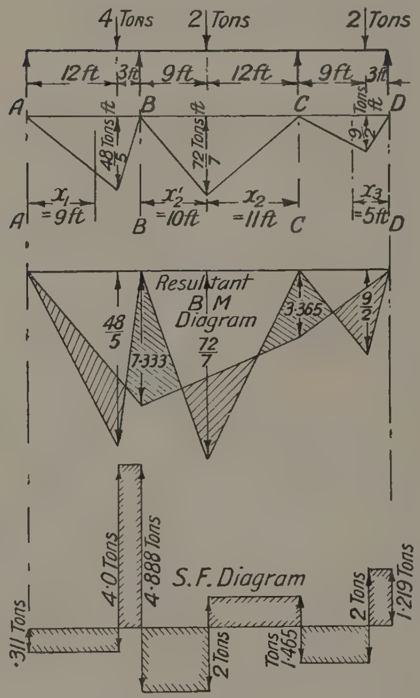

FIG. 57

Moments of forces about B
(right-hand side)

$$(R_c \times 21) + (R_d \times 33) + M_b = (2 \times 9) + (2 \times 30)$$

$$\therefore \quad R_c = \frac{78 - 47{\cdot}56}{21} = \underline{1{\cdot}45} \text{ tons.}$$

Moments of forces about C
(left-hand side)

$$(R_a \times 36) + (R_b \times 21) + M_c = (2 \times 12) + (4 \times 24)$$

$$\therefore \quad R_b = \frac{120 - 14{\cdot}561}{21} = \underline{5{\cdot}02} \text{ tons.}$$

MATHEMATICS FOR ENGINEERS

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LESSON XI

DETERMINATION OF LAWS

In the last lesson we showed how to obtain the equation representing the law connecting two quantities when the corresponding graph was a straight line. With the typical equation

$$y = mx + c \quad . \quad . \quad . \quad . \quad . \quad (1)$$

m was determined by the gradient of the line, but *c* was found by inspection, as the value of y when $x = 0$. It will often happen, however, that it is not convenient to have $x = 0$ on the graph, and in such a case we must obtain the value of *c* by some further calculation. If *m* has been found, we can use the values of *y* and *x* for any point on the line, inserting them in (1) and solving for *c*. Alternatively, we may take the values of *x* and *y* for any two points on the line, insert them in (1), thus obtaining two equations for *m* and *c*, and solve them by the methods of Lesson VIII.

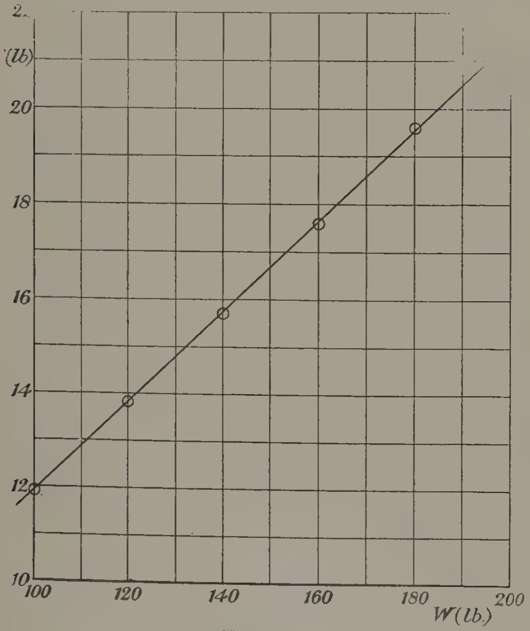


FIG. 22

As an instance, take the results of experiments on a screw-jack—

Load, *W* lb. .	100	120	140	160	180	200
Effort, *P* lb. .	11·8	13·8	15·7	17·6	19·6	21·5

Here, if we want *P* in terms of *W*, to plot *W* from 0 to 200 means half the scale, and consequently half the accuracy, of plotting 100 to 200 only. The graph is given in Fig. 22. From it we see that when *W* is 100, *P* (from the *line*) is 11·94 ,and when *W* is 200, *P* is 21·44. So

$$m = \frac{21{\cdot}44 - 11{\cdot}94}{200 - 100} = \frac{9{\cdot}5}{100} = \underline{0{\cdot}095}$$

Now use the first of these points to find *c*, inserting the values in

$$P = mW + c \quad . \quad . \quad . \quad . \quad (2)$$

obtaining

$$11{\cdot}94 = 0{\cdot}095 \times 100 + c$$

whence

$$c = 11{\cdot}94 - 9{\cdot}5 = \underline{2{\cdot}44}.$$

So the law is

$$\underline{P = 0{\cdot}095W + 2{\cdot}44.}$$

Alternatively, substituting both pairs of values in (2), we have

$$11{\cdot}94 = 100m + c$$
$$21{\cdot}44 = 200m + c.$$

Subtracting

$$9{\cdot}50 = 100m$$

whence

$$m = \underline{0{\cdot}095}, \text{ as before.}$$

The value of *c* is now found as above.

Power Laws. While the " straight line law " we have just been studying is very common, yet there are many cases of practical occurrence where the graph is *not* straight, but where we still desire a formula to represent it. Provided its shape justifies it—and a little experience will enable us to recognize when this is so—the next trial is usually a " Power Law," i.e. a formula of the type

$$y = cx^n \quad . \quad . \quad . \quad (3)$$

where c and n are to be determined to fit the curve as well as possible. In Lesson VIII we found that it was possible to determine unknown indices by the use of logarithms, and we use the same device here. Taking logarithms of both sides of (3), we have

$$\log. y = n \log. x + \log. c \quad . \quad . \quad . \quad (3a)$$

If we write $Y = \log. y$, $X = \log. x$, $C = \log. c$ we have

$$Y = nX + C \quad . \quad . \quad . \quad . \quad (3b)$$

Now (3b) is of exactly the same *form* as (1), so that plotting Y and X (that is, $\log. y$ and $\log. x$) will give, if (3) is correct, a straight line. Moreover, the gradient of this line is, as we see by comparing with (1), n. Thus, the test for a "power law" is to plot logarithms of the two variables, when a straight line indicates the existence of such a law, and its gradient is the index.

EXAMPLE. In calibrating a weir to measure flow of water, the following were obtained experimentally.—

Head H, (in.) . . .	2	6	10	15
Discharge Q (cub. ft./min.) .	5·93	30·9	66·3	122

We require a formula for Q in terms of H.

The small $Q-H$ graph, Fig. 23a, shows that a power law is probable, so we proceed to take logs. The new table is

Log. H . .	·3010	·7781	1·0000	1·1761
Log. Q . .	·7731	1·4900	1·8215	2·0864

Plotting these, in Fig. 23b, we obtain a practically straight line. From this, when $\log. H = 0·3$, $\log. Q = 0·764$, and when $\log. H = 1·1$, $\log Q = 1·966$. Hence, the gradient

$$n = \frac{1·966 - 0·764}{1·1 - 0·3} = \frac{1·202}{0·8} = \underline{1·50.}$$

Using this value of n and the values at the first of the above points,

$$0·764 = 0·3 \times 1·50 + \log. c$$

or, $\quad \log. c = 0·764 - 0·450 = 0·314$

so that $\quad c = \underline{2·06.}$

The required formula is thus,

$$\underline{Q = 2·06\, H^{1.5}.}$$

EXERCISE No. 18

1. If h is the loss of head due to friction in a pipe, and v the velocity of the water, find a formula of the type $h = Cv^n$ from the following—

v (ft./sec.) .	4	6	8	10
h (ft.) . .	·0143	·0305	·0512	·0765

2. The horse-power, H, required to propel a boat at v ft./sec. was found to be as follows—

H	1054	2125	3850
v	20·1	24·9	30·2

Find H in terms of v, assuming a formula of the form $H = Cv^n$.

3. By plotting *log p* against *log. v*, find the most probable values of n and c if the following are to be represented by the law, $pv^n = c$.

p . .	100	91·3	84·5	78·2	72·4	67·9
v . .	3·0	3·2	3·4	3·6	3·8	4·0

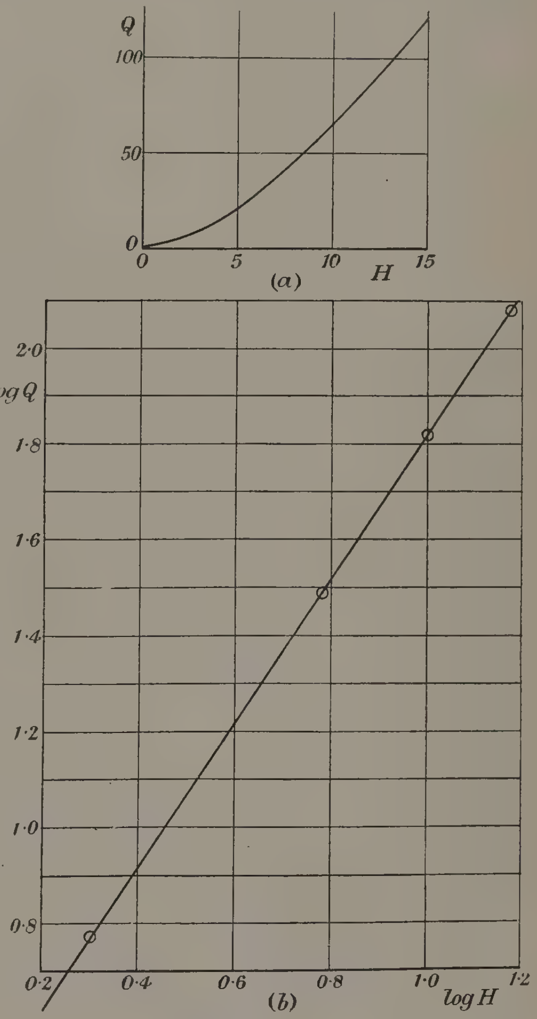


FIG. 23

AREAS UNDER CURVES

There are cases where the area bounded by a curve is desired, for its own sake, such as the area of the water line section of a ship, and there are others in which the area *represents* something, an instance of this being the area enclosed by an indicator diagram, which represents the work done in a cycle, to a scale of so many ft.-lb. per sq. in., the actual number depending upon the stiffness of the indicator spring and the strokes of the piston and indicator. True, the indicator diagram is more often used to calculate the average pressure, and the work then found by multiplying this by the stroke, but the average pressure is the average height of the diagram, and the stroke is represented by its length, so that it is the area of the diagram that represents the work done. We will illustrate the methods of finding the area, as well as the fact that in many cases the area has a meaning, by working a few examples.

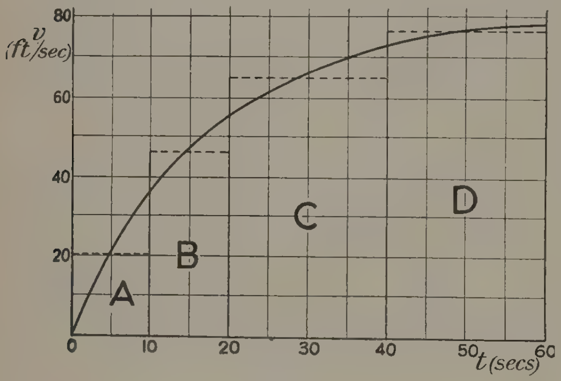


Fig. 24

The speed, v, reduced to ft./sec., of a train, t secs. after starting is

t	.	0	10	20	30	40	50	60
v	.	0	36·5	55·4	66·0	72·8	76·4	78·0

The graph is drawn in Fig. 24, and we shall use it to find the distance travelled in the 60 secs. To see that it is the area under the curve that represents the distance, consider Fig. 25, a sketch of the same. Take two ordinates, AB and CD, with horizontals BF and ED. The width, AC, of the strip represents a small interval of time, while the heights, AB and CD represent the speeds at the beginning and end of the interval, respectively. The area of the rectangle $ABFC$ is $AC.AB$, or (interval of time) × (initial speed). Now this is the distance that would be travelled in the interval, at the initial speed, so that the area of the rectangle represents this distance. Also, in a similar way, the area of the rectangle $AEDC$, which is $AC.CD$, represents the distance that would be travelled in the interval at the final speed. Now the actual distance travelled is between these two, and the area under the curve is the only area that always, wherever we take A and C, lies between the two rectangles, so that it must represent the actual distance. To find the area in Fig. 24, we divide it into strips. More strips mean greater accuracy, but it is unwise to use too many. Make them narrow where the height changes quickly, but broader where it changes slowly. We take four portions, A, B, C, and D, by ordinates at 10, 20, 40, and 60. We now estimate the average height of each strip, drawing in the horizontals as shown. To calculate the area, it is best to work in a tabular form, as below—

Strip	Average Height	Base	Area
A	20·5	10	20·5 × 10 = 205
B	46·2	10	46·2 × 10 = 462
C	65·0	20	65·0 × 20 = 1,300
D	76·0	20	76·0 × 20 = 1,520
			Total 3,487

The distance covered is then 3,487 ft. In spite of the apparent roughness of the estimation of the average height, this method yields quite good results, not differing by more than 2 or 3 per cent in a class of students.

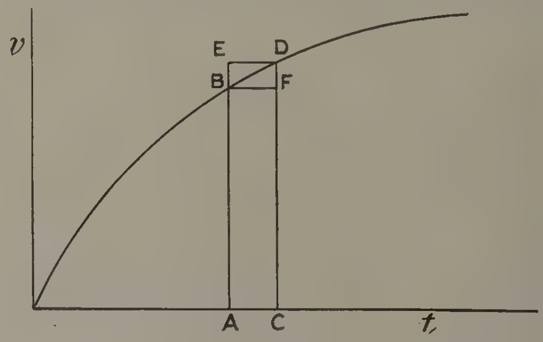


Fig. 25

Simpson's Rule. The above method has many advantages over the "Mid-Ordinate" and the "Trapezoidal" rules often given, since they make no attempt to take into account the curvature. Simpson's rule, however, does, and gives very accurate results. To use it, we must divide the base into an *even* number of strips, all of the same width, and thus an *odd* number of ordinates. Calling the common width of strip h, the area is given by

$$\tfrac{1}{3}h \times \text{sum of} \left\{\begin{array}{l}\text{first and last}\\ \text{4 times the even}\\ \text{twice the other odd}\end{array}\right\} \text{ordinates.}$$

The calculation is conveniently tabulated, as shown below, where it is used to calculate the distance represented by the area in Fig. 24, using the six strips, and the ordinates from the Table given above.

(1) No. of Ordinate	(2) Ordinate	(3) Simpson's Multiplier	(4) (2) × (3)
1	0	1	0
2	36·5	4	146·0
3	55·4	2	110·8
4	66·0	4	264·0
5	72·8	2	145·6
6	76·4	4	305·6
7	78·0	1	78·0
		Total	1,050·0

$$h = 10$$

$$\text{area} = \frac{10}{3} \times 1050 = \underline{3500.}$$

It will be seen that the estimation method agrees very well indeed with this, probably more accurate, result.

For all but the most accurate work, the estimation method is sufficiently good. Moreover, unless the ordinates can be *calculated* at desired positions, and the base is conveniently divisible into an even number of equal parts, Simpson's rule is not convenient, nor is it more accurate. Both, however, should be known.

There are many other instances in which the area under a curve has a meaning, and the meaning, if any, can be found by considering a narrow strip in the same way as we have done above for Fig. 25. It is, of course, important that the right quantity be plotted horizontally in these cases. Another way of considering the matter is to think, in the above example, that the distance gone is given by (*total* time) × (*average* speed)—the other possibility (total velocity) × (average time) is not sense—and as the area is (base) × (average height), we must plot *time* horizontally. A few other cases when an area has a meaning are—

Quantities Plotted		Meaning of Area
Horizontally	Vertically	
Time	Acceleration	Speed
Displacement	Force	Work done
Height, depth	Cross-section	Volume
Volume	Pressure	Work done
Time	Force	Momentum

EXERCISE No. 19

1. The acceleration of a piston over an interval of time is as follows—

Time (sec.) .	0	0·015	0·045	0·075	0·105	0·135
Acc. (ft./sec².) .	90	87·6	77·9	57·9	30·8	0

If it starts from rest, find its final velocity.

2. The area of cross-section of a reservoir, at various depths, is given by the following table—

Depth water (ft.)	0	5	10	20	30	40
Area (sq yd.) .	0	1,850	2,600	3,250	3,950	5,300

Find the total contents in gallons, when full, and also the amount which leaves the reservoir when the level falls from 40 to 25 ft. (1 cub. ft. = 6·25 gal.)

3. The pressure of the steam in an engine cylinder, for various volumes, is given by—

Pressure (lb./sq. in.) .	200	100	66·7	50	40
Volume (cub. ft.) . .	0·1	0·2	0·3	0·4	0·5

Find, in ft.-lb., the total work done in expansion from 0·1 to 0·5 cub. ft.

ANSWERS TO EXERCISE No. 17

1. 287, 572, 766, 6·30, 8·77, 9·44.
2. (*a*) 4·83, −0·83; (*b*) 1·54, 0·35, −1·88; (*c*) $x = 2$, $y = 3$.
3. $B = 0{\cdot}97I - 15{\cdot}8$.
4. $W = 15K + 21{,}500$.

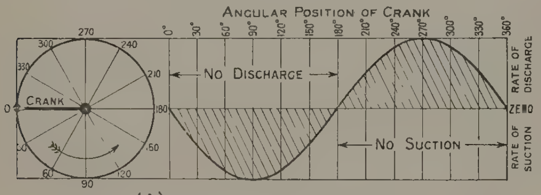
PUMPS: CONSTRUCTION AND MAINTENANCE

By Owen A. Price, M.I.Mech.E.

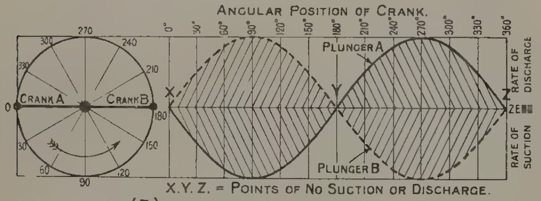
LESSON VI

GENERAL ARRANGEMENT

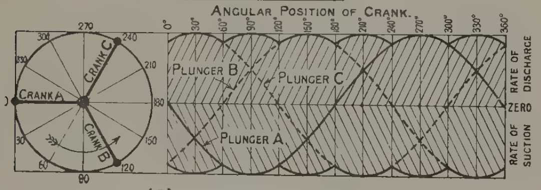
Single and Multiple Units. A pumping unit commonly consists of one, two, or three working barrels, the choice depending upon the importance of securing a uniform driving torque and a regular rate of discharge free from serious oscillation.



(A) Single Ram Pump.



(B) Double Ram Pump.



(C) Treble Ram Pump.

Fig. 21. Rates of Pumping

The variation in the rate of discharge for single, double, and treble ram pumps is shown in Fig. 21, *A*, *B*, and *C*.

Horizontal and Vertical Designs. The vertical form (Figs. 3 and 6) has the advantage in floor space, but the disadvantages that the crankshaft carrying the rotating masses, and transmitting the pumping torque, is high above the foundation, and also that the general centre of gravity is high. The horizontal type (Fig. 11) has a low centre of gravity and great rigidity. In vertical arrangements the whole pumping load is thrown on the crankshaft-bearing caps and bolts, whilst in the horizontal form this load is more securely borne by the bedplate itself.

The popularity of the vertical pump, however, is mainly due to the advantageous arrangement of the plunger and barrel, by which wear takes place evenly all round, and grit is less likely to reach the glands and to affect the working surface of the barrel. With horizontal pumps the weight of the plunger or piston and rod, etc., causes the wear on glands, bushes, barrel, and plunger to take place in a downward direction, and grit by settling to the bottom contributes to increased wear in the same direction.

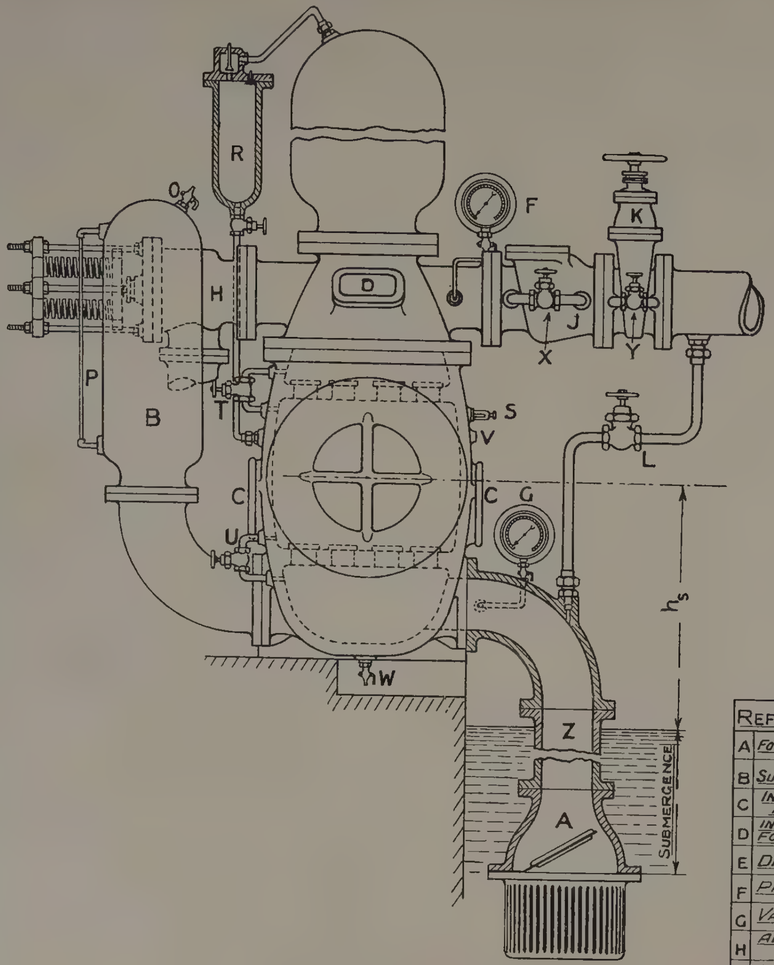
Driving Means. The method of driving the pump will be determined by the conditions of each case. Belt drives have the great advantage of flexibility, and are easy on the pump because the belt readily absorbs shocks. Spur gearing is frequently necessary though not an ideal drive, because it is harsh on the pump and frequently noisy. Herringbone gears and raw hide or flexible pinions are often adopted to lessen the objectionable features, but a wise policy is to reduce gearing to the bare minimum.

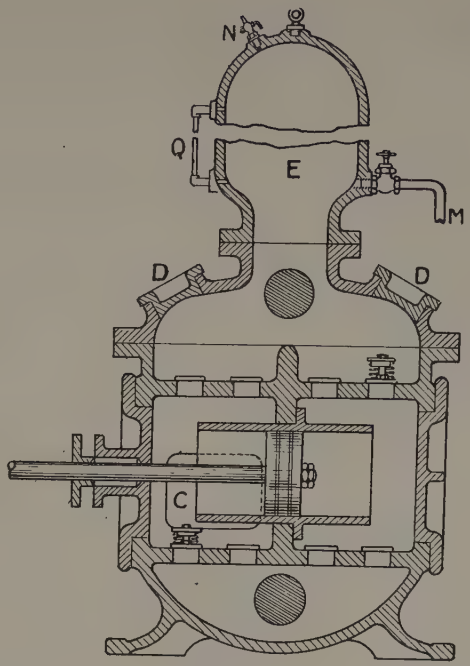
ACCESSORIES AND FITTINGS

To complete a pumping equipment various accessories are required in the pipe line and certain fittings for the pump. Fig. 22 has been drawn to show the location of the various items, some only of which would be provided with a small pump and most of which might appear on a large pumping unit.

A *foot valve* (*A*) is necessary in cases of high suction lift, or when the pump is designed with a large clearance volume (see Lesson III, "Clearance"). Before starting, the pump may then be charged with water, either by means of a *priming connection L*, or by opening the *bypasses T*, *U*, *X*, and *Y* across the various valves.

The functions of a *suction air vessel* (*B*) are to maintain a uniform velocity of flow in the suction pipe, so avoiding water-hammer, cavitation, or inertia losses, and, also to reduce shock when the suction valves close. To fulfil its duty as a velocity equalizer, the suction air vessel should be in direct line with the oncoming



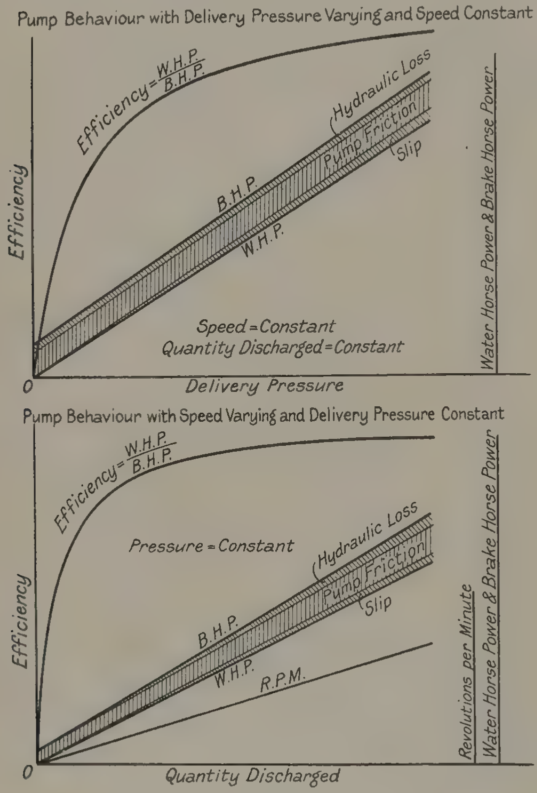


REFERENCE DIAGRAM FOR PUMP ACCESSORIES AND FITTINGS

A	Foot Valve & Strainer	K	Sluice Valve	T	Bye-Pass Across Delivery Valves
B	Suction Air Vessel	L	Priming Connection	U	Bye-Pass Across Suction Valves
C	Inspection Door For Suction Valves	M	Air Escape & Overflow When Priming	V	Indicator Plug
D	Inspection Door For Delivery Valves	N	Air Cock For Delivery Air Vessel	W	Drain Cock
E	Delivery Air Vessel	O	Air Cock For Suction Air Vessel	X	Bye-Pass Across Reflux Valve
F	Pressure Gauge	P	Gauge Glass For Suction Air Vessel	Y	Bye-Pass Across Sluice Valve
G	Vacuum Gauge	Q	Gauge Glass For Delivery Air Vessel	Z	Suction Pipe
H	Alleviator or Relief Valve	R	Inspirator or Air Vessel Charger		
J	Reflux Valve	S	Snuffle or Snifting Valve		

FIG. 22. ACCESSORIES AND FITTINGS

water from the suction pipe, and so arranged that it can both receive and discharge water with the minimum deflection of the stream entering the pump. As a valve shock reducer the air vessel is ineffective unless quite close under the suction valves. When fitted to commercial pumps the volume of a suction air vessel is about three to five times the pump displacement per stroke. Such a volume is suitable for normal running speeds on low suction heads up to 15 ft. For 20 ft. suction head 10 times, and for 25 ft. suction head, fifteen times the plunger displacement should be provided. Where the suction pipe is long or contains several bends, these volumes should be increased by 20 to 30 per cent.


Pump Behaviour with Delivery Pressure Varying and Speed Constant

Pump Behaviour with Speed Varying and Delivery Pressure Constant

FIG. 23. TEST PERFORMANCE

The *delivery air vessel E* serves the delivery valves and delivery pipe in the same manner that the suction air vessel serves its valves and pipe, and the remarks made concerning the one apply to the other. The head on the delivery side, however, is, as a rule, much greater than the head (negative) on the suction side. The capacity of a delivery air vessel, therefore, is usually greater than a suction air vessel, volumes from five to thirty times the pump displacement being common, and in special cases even a hundred times.

An approximate rule is

$$\frac{V}{D} = P \times k$$

where V = volume of air vessel
D = displacement of pump
P = the delivery pressure in pounds per square inch
k = ·10 for a single-ram pump
·07 for a double-ram, or a double-acting pump
·03 for a triple-ram pump.

Where very long pumping mains occur it is usual to fit air vessels over each set of delivery valves, and a much larger air vessel on the pipe to deal with fluctuations and surge in the main. For very heavy pressures a powerful spring-loaded *pressure equalizer* is substituted for the air vessel.

The *alleviator or relief valve* is the safety valve for avoiding damage in the event of excessive pressure, due, for example, to obstruction in the mains or the closing of a sluice valve during pumping.

Both a *reflux valve J* and a *sluice valve K* are shown in Fig. 22. Either would serve to isolate the pump when necessary. Where a reflux valve is used it is advisable to fit an air vessel on the outlet side to serve the pipe line.

The *overflow pipe M* is opened when priming, and is intended to ensure that the air vessel is not filled with water.

There is always some air dissolved in natural water and, as this air readily escapes when the water is subjected to the vacuum of the suction pipe, a suction air vessel is not liable to become waterlogged. A delivery air vessel is less fortunately circumstanced because water under pressure has the property of absorbing an increased quantity of air, consequently the delivery air vessel is continually losing air by solution with the water. Unless replenished with a fresh supply, therefore, the vessel becomes waterlogged and useless. The *inspirator R* is one of the means employed for introducing air to the delivery air vessel. By opening the stop-cock communicating with the pump barrel, air is sucked in through the small atmospheric valve during the suction stroke of the main pump and is forced into the air vessel during the delivery stroke.

The *snuffle or snifting valve S* serves a similar purpose, though less efficiently, because the admission of air to the working barrel impairs the volumetric efficiency of the pump. Admitting air under the delivery valves by a snifting valve is also a makeshift for silencing a noisy pump.

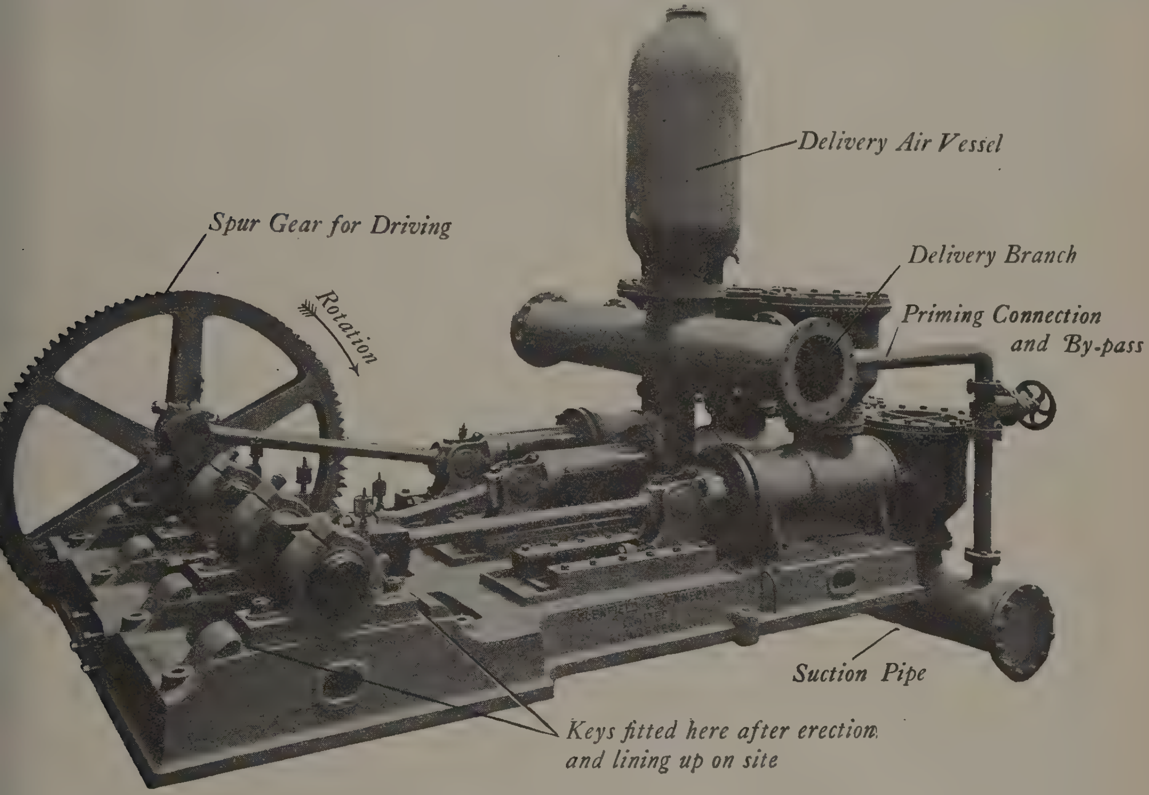

Delivery Air Vessel
Spur Gear for Driving
Rotation
Delivery Branch
Priming Connection and By-pass
Suction Pipe
Keys fitted here after erection and lining up on site

FIG. 24. HORIZONTAL THREE-THROW RAM PUMP

By-pass valves T, U, X, and *Y* serve either for priming, for relieving a pump of pressure when starting up, or for draining the various parts.

Suction Pipe. *Z.* The pipe end, whether bell-mouthed or with foot valve, should be submerged at least 3 ft. below the lowest water level, and ample free space provided all round it. The suction pipe must be as short and straight as possible with the minimum number of *easy* bends, and any horizontal portions should be laid at a small gradient so that the pipe rises all the way to the pump, thereby avoiding air pockets.

Preferably the vertical suction height should not exceed 15 to 18 ft., though 22 ft. (exceptionally 25 ft.) may be considered a maximum practical limit if the pipe is short and direct.

Wherever abnormal conditions occur, such as long or constricted suction pipes, high pump speed, great suction lift, etc., the suction pipe conditions must be carefully calculated to ascertain if the water is likely to separate from the plunger. The height of the water barometer (say, 34 ft.) sets a definite limit to the head available for accelerating and forcing the suction water into the pump barrel in sufficient volume.

Using feet units throughout and calling h_s the vertical suction height and h_f the friction head, the net accelerating head acting on the water surface is $(34 - h_s - h_f)$, and this will produce an acceleration $= \frac{(34 - h_s - h_f)g}{l}$ where l is the length of oscillating water column measured along the suction pipe to the plunger.

The crank pin velocity $u = \frac{2\pi r N}{60}$ ft. per second, where N = revolutions per minute, and r = the crank radius in feet. Also, the acceleration of the plunger at the beginning of the stroke $= \frac{u^2}{r}$, and if A = the area of the plunger and a = the area of the suction pipe, then the corresponding acceleration of the water in the suction pipe $= \frac{A}{a} \times \frac{u^2}{r}$.

For the water to follow the plunger, therefore

$$\frac{(34 - h_s - h_f)g}{l} \text{ must be } \left\{ \begin{matrix} \text{equal to or} \\ \text{greater than} \end{matrix} \right\} \frac{A}{a} \times \frac{u^2}{r}$$

Inserting the value of u above, and reducing

$$N_{(max)} = 54 \sqrt{\frac{(34 - h_s - h_f)}{lr} \times \frac{a}{A}}$$

or, correcting for the finite length of the connecting rod where

n = the ratio $\frac{\text{connecting rod length}}{\text{crank radius } r}$, this becomes

$$N_{(max)} = 54 \sqrt{\frac{34 - h_s - h_f}{lr(1 + \frac{1}{n})} \times \frac{a}{A}}$$

Or, the maximum suction lift

$$h_{s(max)} = 34 - h_f - \left\{ \frac{N^2}{2940} \times lr\left(1 + \frac{1}{n}\right) \frac{A}{a} \right\}$$

For reliable operation in practice the actual suction lift would be made 5 to 10 ft. less than $h_{s(max)}$ found above.

These exceedingly important relations indicate when the diameter of the suction pipe should be increased, when a suction air vessel is necessary for the purpose of shortening the oscillating water column length l, and enable other important proportions to be studied with regard to their practical effect.

TEST PERFORMANCE

Practical tests are usually carried out under the particular conditions for which the pump is intended to work. However, it is instructive to study the behaviour of a pump at varying conditions of speed and pressure. Typical performance curves are shown in Fig. 23, the *speed* remaining constant in the one case and the *delivery pressure* constant in the other. When a certain amount of data is available from similar types of pump it is a simple matter to estimate such diagrams by building up the components of the lost power on the *WHP* line in the manner indicated on Fig. 23.

PRACTICAL EXAMPLE

A typical horizontal power-driven pump is shown in the photograph, Fig. 24. This has three plungers, each $11\frac{1}{2}$ in. diameter by 24 in. stroke, and delivers 834 imp. gals. per minute against 167 ft. head when running at 35 revs. per minute.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON X

Proportion in Drawing. We have, in a previous lesson, referred to the use of the *reference drawing* as a guide to machine construction, and as an adjunct to the *reference drawing* we have various rules of proportion.

The production of a machine drawing consists, in a great measure, of the building of a number of parts or sections of parts in what approximates to an established and rational proportion. Further, no small part of the art of machine construction consists in discriminating between that portion of the machine part which, for reasonable strength, requires careful proportion, and that part which, for practical reasons, obtains a proportion of more than average strength.

The common case in which rules of proportion are used is where the dimension of one part or section is determined by a more or less lengthy calculation, and the remaining dimensions found from simple proportion to the first fixed dimension. Such an example is given in Fig. 1, which shows two steel shafts connected by couplings forged solid with the shafts. We may assume the diameter of the shaft to be given, it being our problem to obtain suitable dimensions to allow a working drawing of the couplings to be made.

First, in finding the diameter of the coupling bolts, we have, where

n = number of bolts in coupling.
d = diameter of bolts in coupling in inches.
D = diameter of shafts in inches.
R = pitch radius of bolts in inches.
f_s = shear stress induced in shaft.
f_b = ,, ,, ,, bolts.

Then, when any power is being transmitted between the shafts

Moment of resistance of bolts
= twisting moment in shaft.

$$n \times \frac{\pi d^2}{4} \times R \times f_b = \frac{\pi D^3}{16} f_s \quad . \quad . \quad . \quad (1)$$

making $f_b = \cdot 8 f_s$

,, $R = \cdot 8D$

and circular pitch of bolts $= 3d$

$$n = \frac{2\pi R}{3d} = \frac{2\pi \times \cdot 8D}{3d}$$

substituting from (1)

$$\frac{2\pi \times \cdot 8D}{3d} \times \frac{\pi d^2}{4} \times \cdot 8D \times \cdot 8 f_s = \frac{\pi D^3}{16} f_s$$

$$d = \frac{D}{4 \cdot 27}, \text{ say, } \frac{D}{4} \text{ approx.}$$

To determine the thickness of the flange t ve have, where f_c = crushing stress induced in he bolts,

$$f_c dt = \frac{\pi d^2}{4} f_b$$

nd making $fc = 2f_b$

$$2dt = \cdot 7854 d^2$$
$$t = \cdot 393 d$$
$$t = \cdot 098 D.$$

Whilst the above proportion of flange thickness is sufficient to provide an average strength o this section, it is necessary to increase the limension in order to maintain the alignment f the coupling bolt. The above is one of many ases in machine construction where rigidity and not strength determines the proportion of a machine part. In practice, the thickness of the coupling flange would be made about $\cdot 25D$.

The remaining dimension to be fixed is the diameter of the coupling. For the purpose of strength a diameter very little more than the PCD of the bolts would suffice, and so, in practice, it is the rule to extend the flange diameter just beyond the extreme radius of the coupling bolt nut.

Tabulated Proportions. Where machine parts of similar design but varying size are in frequent demand, then a tabulated list of dimensions based upon known proportions may be found economical, both with regard to draughtsman's time and drawing space.

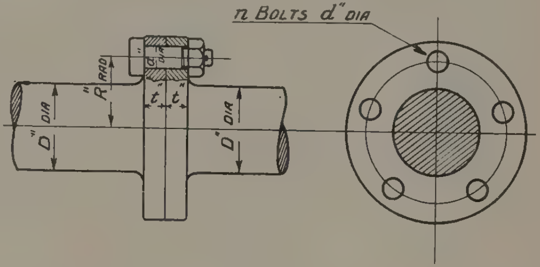


FIG. 1

Suppose drawings for cast-iron flanged couplings are frequently called for, then, having found suitable proportions for a coupling to fit a certain diameter of shaft, an extension of the rules adopted will, with very little extra work, provide dimensions for couplings to fit shafts of other diameters. Suppose the type of coupling to consist of two cast-iron flanges keyed to the adjoining shafts and connected by bolts, as shown in Fig. 2, then we may proceed as follows—

Commencing with the proportion of the key, we have, where

f_s = shear stress induced in the key
f = ,, ,, ,, shaft
f_c = crushing ,, ,, key

and referring to Fig. 2

$$EJf_s \frac{D}{2} = \frac{\pi D^3}{16} f$$

making $f_s = f$

$$EJ = \frac{\pi D^2}{8}$$

E and J being variables it is necessary to fix one, and making $J = \frac{D}{4}$ we have

$$\frac{ED}{4} = \frac{\pi D^2}{8}$$

$$E = \frac{\pi D}{2} = 1{\cdot}57D.$$

To determine a suitable value for the depth of key we have to observe that the key, if well fitted, is so restrained against distortion that with orthodox proportions loading to the limit would result in failure by shearing the key. We therefore fix the depth of key with a view to maintaining a good fit, rather than to avoid failure by crushing. The ratio of crushing stress to shear stress commonly adopted in practice is $\frac{fc}{fs} = 4$. Accepting this ratio, we have

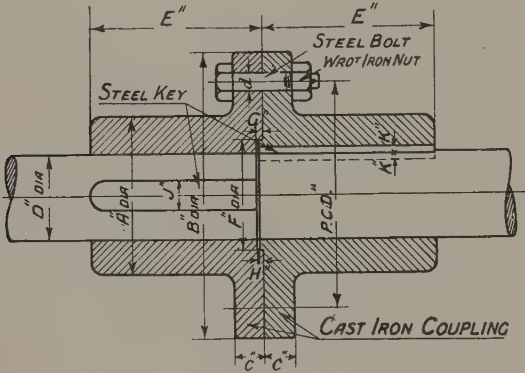
$$EK\ \frac{D}{2} f_c = EJ\ \frac{D}{2} f_s$$

$$K \times 4 \times f_s = J \times f_s$$

$$K = \frac{J}{4}.$$

We may add that for shafts of large diameter, carrying wheels or discs, which may be subjected to rapid changes of torque, the ratio $\frac{f_c}{f_s}$ is often reduced to a value of 3. When this smaller ratio is adopted, the value of E is made correspondingly larger, in order to avoid an undue reduction of shaft section.

In finding a value for A we may consider the coupling, apart from the flanges, to consist of a hollow circular section of internal diameter D and external diameter A. To provide equal resistance to torque when transmitting power, the dimension A need not exceed $1{\cdot}22D$. For practical reasons, and because the section is subject to bursting stresses of unknown magnitude, the ratio of outside to inside diameter is made much greater than 1·22. For a cast-iron section adjoining a steel shaft a ratio of 1·8 provides a better proportion. For the remaining proportions we may proceed as in the case of the solid flanged coupling in Fig. 1, and obtain a ratio of $PCD = 2{\cdot}6\ D$ and $d = \frac{D}{5}$.



Particulars of Cast Iron Couplings.

D	A	B	C	E	F	G	H	J	K	d	P.C.D.	No Bolts
2	$3\frac{5}{8}$	7	$\frac{3}{4}$	$3\frac{1}{2}$	3	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{5}{8}$	$\frac{5}{32}$	$\frac{1}{2}$	$5\frac{3}{8}$	4
$2\frac{1}{2}$	$4\frac{1}{2}$	$8\frac{1}{4}$	1	$4\frac{1}{2}$	$3\frac{1}{2}$	"	"	$\frac{3}{4}$	$\frac{3}{16}$	$\frac{5}{8}$	$6\frac{1}{2}$	4
3	$5\frac{3}{8}$	10	$1\frac{1}{8}$	$5\frac{1}{2}$	4	"	"	$\frac{7}{8}$	$\frac{7}{32}$	$\frac{3}{4}$	$7\frac{3}{4}$	4
$3\frac{1}{2}$	$6\frac{1}{4}$	$11\frac{1}{4}$	$1\frac{3}{8}$	$6\frac{1}{4}$	$4\frac{1}{2}$	"	"	1	$\frac{1}{4}$	$\frac{3}{4}$	9	5
4	$7\frac{1}{4}$	13	$1\frac{1}{2}$	7	5	"	"	$1\frac{1}{8}$	$\frac{9}{32}$	$\frac{7}{8}$	$10\frac{1}{2}$	5
$4\frac{1}{2}$	8	$14\frac{1}{4}$	$1\frac{3}{4}$	8	$5\frac{1}{2}$	"	"	$1\frac{1}{4}$	$\frac{5}{16}$	$\frac{7}{8}$	$11\frac{3}{4}$	6
5	9	16	$1\frac{7}{8}$	9	6	"	"	$1\frac{3}{8}$	$\frac{11}{32}$	1	13	6

Fig. 2

The spigot (Fig. 2) is designed to effect the alignment of the couplings during machining and after fitting to the shafts. A drawing may now be made, as shown in Fig. 2, the dimensions being replaced with letters, and a table of dimensions under corresponding notation made for varying diameters of shaft. In using this table for a shaft of intermediate diameter the tabulated dimensions can be interpolated.

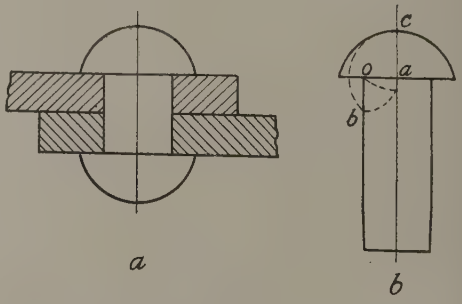


Fig. 3

Riveted Joint. A further example of proportion in drawing is provided in the case of a single riveted lap joint, shown in Fig. 5.

The proportions of the rivet heads are commonly standardized and, as a rule, made by specialists in the manufacture of rivets and bolts. Consequently, in machine drawing, the form and proportions of rivet heads are required solely to allow a reasonably true outline of the rivet to be shown. A rapid and sufficiently accurate method of drawing two types of rivet heads is shown in Figs. 3 and 4.

Fig. 3 (*a*) shows a snap-head rivet, and Fig. 3 (*b*) shows the method of obtaining the depth and radius of the head. First drawing lines representing the body of the rivet, and from the point *O* describing an arc *ab* of radius equal to the radius of the rivet cutting the rivet line at *b*, then the chord *ab* gives the depth of the head *ac*. Further, the chord *oc* gives the radius of the head.

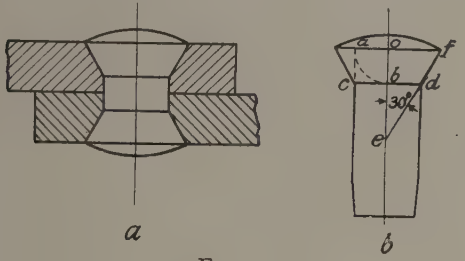


FIG. 4

Fig. 4 shows a countersunk head type of rivet and, to construct an outline of the same, we again draw lines representing the body of the rivet. Next, with *o* as centre and *oa* as radius, describe the arc *ab*, cutting the centre line at *b*. The distance *ob* is the depth of plate countersunk. Proceeding, draw lines at an angle of 30° to the rivet centre line cutting the lines of diameter at *c* and *d*. Then *ef* is the radius of the rivet head.

The basis of an economical design is to endeavour so to proportion the parts that the stress induced in each section bears a common ratio to its ultimate stress. In practice, other considerations modify this proportion, but, as a rule, one ought to commence with the proportions giving equal strength and only deviate from these for practical considerations. For example, in Fig. 5 we have a single riveted lap joint. Analysing the stresses induced in the joint, and taking a portion of the joint over a pitch length, we may consider three sections of the joint.

At section *ab* the forces *F* tend to shear the rivet.

If f_s = shear stress induced in the rivet
f_c = crushing stress induced in the rivet
f_t = tensile stress induced in the plate
then $F = \cdot 7854 d^2 f_s$.

At the section *ce* the same forces tend to tear the plate between the rivets, and $F = (p-d)t.f_t$.

At the section *fg* the forces *F* tend to crush the rivet, and $F = d.tf_c$.

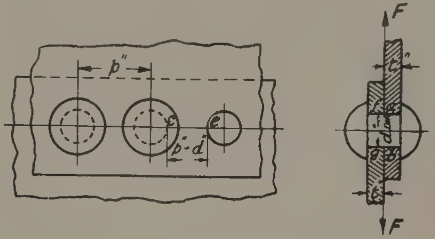


FIG. 5

Then, for equal strength,

$$\cdot 7854 d^2 f_s = (p-d)t.f_t$$

making $f_s = \cdot 8 f_t$

$$p - d = \frac{\cdot 7854 d^2 \times \cdot 8}{t}$$

$$p = \frac{\cdot 63 d^2}{t} + d.$$

Also $\cdot 7854 d^2 f_s = d.t.fc$

making $f_c = \dfrac{7}{4} f_s$

$$d^2 = \frac{dt}{\cdot 7854} \times \frac{7}{4} = 2\cdot 22 d.t$$

$$d = 2\cdot 22t, \text{ say, } 2t.$$

EXERCISE

(*a*) Using the table of proportions given in Fig. 2, make a working drawing of a cast-iron coupling for a shaft $3\frac{3}{4}$ in. diameter.

(*b*) Find the pitch and diameter of rivets for a single-riveted lap joint, the plates being $\frac{1}{2}$ in. thick and the rivets having snap heads.

POWER TRANSMISSION BY BELTING

By W. G. Dunkley, B.Sc.

LESSON VI

STRENGTH OF BELTING AND ALLOWABLE LOADS

We have so far seen that we have three tensions to consider, the initial, tight, and slack side tensions. The maximum stress in the belt is not determined by the driving tension, which is the difference between the tight and slack side tensions, but by the actual value of the tight side tension. Just as in a steel bar, we compare our load with the actual or ultimate breaking load and so determine our factor of safety, so with a belt we must calculate our factor of safety by comparing the actual maximum tension with the breaking strength of the belt, although our factor of safety will not be a steady factor, but, as we have seen, one which is gradually diminishing in value as time goes on.

Ultimate Strength of Belting. We shall now quote some figures relating to the ultimate strength of belting. The results of tests made by Kirkaldy show that *ordinary tanned leather* has an ultimate strength in pounds per square inch varying from 4,800 lb. for a belt of 0·2 in. thick to 2,500 lb. for a belt 0·5 in. thick.

For *solid woven cotton* the ultimate strength for belts 0·26 in. thick varies from 8,800 to 6,800 lb. per sq. in. For *folded stitched cotton* 0·2 to 0·4 in. thick the ultimate stress varied from 7,700 to 5,800 lb. per sq. in. For the leather belting, the average breaking stress per square inch for single oak-tanned leather is about 4,000 lb., and for double belting 3,000 lb. It would be reasonable, therefore, to assume that double belting has 75 per cent the strength of single belting per square inch.

The following figures are given by Messrs. Tullis—

1 sq. in. of oak-tanned leather belt stands 5,746 lb.
1 ,, ,, common- ,, ,, ,, 4,900 ,,
1 ,, ,, orange- ,, ,, ,, 8,244 ,,

The orange-tanned belting, besides being stronger than the oak-tanned, is 25 per cent lighter and has 30 per cent less stretch. For *solid woven cotton* belting the following figures are given: *Heavy make*, 2,365 lb. per inch width. *Best ordinary*, 1,820 lb. per inch width.

Allowable Maximum Stress. We are now in a position to consider the values allowable for the maximum stress in the belt. We have already made it clear that we must design our belt from a known maximum stress if we are to avoid excessive stretching. To design on a basis of effective driving load per square inch is to leave oneself in the dark as to the actual maximum stress which exists. We have stated above that single oak-tanned leather belt has an ultimate strength of 4,000 lb. per sq. in. If we assume a nominal factor of safety of 10 to 1, this would give us an allowable maximum stress of 400 lb. per sq. in.

Mr. Wilfred Lewis, who carried out extensive tests on belting, assumes an allowable working stress of 500 lb. per sq. in.

Mr. Taylor, whose tests have previously been considered, concluded that the most economical average maximum stress for double leather belting was 200 to 225 lb. per sq. in. For single leather belting, assuming that double belting is only 75 per cent as strong, we would have an allowable maximum stress of 270 to 300 lb. per sq. in. It is clear that some variation in opinion can exist as to the allowable maximum stress, but it is, of course, obvious that the lower the maximum stress allowed the slower the rate of permanent stretching, the less often the belt has to be attended to, and the longer the life of the belt. So first cost has to be balanced against freedom from trouble and a longer life. Mr. Taylor found that double leather belting treated with care and running night and day at moderate speeds should last for seven years when under an average maximum load of 358 lb. per sq. in., and for eighteen years when under an average maximum load of 174 lb. per sq. in. Mr. Taylor says, that with one portion of a shop belted with a maximum stress in double belts of 358 lb. per sq. in., and another portion belted with only 200 lb. per sq. in., the belts in the former group were a constant source of trouble, while practically no trouble was experienced with the second group. Again the above values are for cemented belts and we must, therefore, adopt lower values for laced or hooked joints. We believe that from the above figures the following values may reasonably be taken as a basis, and increased or decreased somewhat accordingly, as first cost or durability is the governing factor.

Suggested Maximum Stress Values.

Single leather belt	(cemented)	400 lb. per sq. in.
,, ,, ,,	(laced)	320 ,,
Double ,, ,,	(cemented)	300 ,,
,, ,, ,,	(laced)	240 ,,

Tension Due to Centrifugal Force. Now in addition to the tension existing necessary to give the desired frictional grip and effective pull, we have also tension due to the centrifugal force exerted by the belt in passing round the pulley. We shall not explain this centrifugal force beyond the following remarks, with reference to Fig. 7. Suppose the belt is considered divided into sections, then each may be regarded as exerting an outward force as indicated. To balance these outward forces, tensions T_c are induced in the two sides of the belt.

Let V = velocity of the belt in feet per minute.
W = weight per foot per inch of width.
t = thickness of belt in inches.
b = width of belt in inches.

$$\text{Then } T_c = \frac{W.V^2.b.}{3600 \times g}, \text{ where } g = 32{\cdot}2$$

The stress per square inch S, say, due to centrifugal force $= T_c \div bt$.

$$\therefore S = \frac{WV^2}{3600 \times g \times t} \text{ lb. per sq. in.}$$

From the above expression we see that the stress due to centrifugal force increases as the square of the velocity, and that, therefore, the effect becomes more pronounced as the speed increases. It will also be seen that the stress due to centrifugal force increases as the weight of the belt per foot increases for a given thickness. If the weight per foot per inch width is not known, a reasonable value to take for W may be calculated by taking the weight of 1 ft. of leather, 1 sq. in. section to be 0·5 lb.

EXAMPLE 9. Suppose a belt runs at 4,000 ft. per minute. What is the stress due to centrifugal force assuming a belt $\frac{3}{16}$ in. thick?

We have $W = \frac{3}{16} \times 0{\cdot}5 = 0{\cdot}094$ lb. per ft. in width.

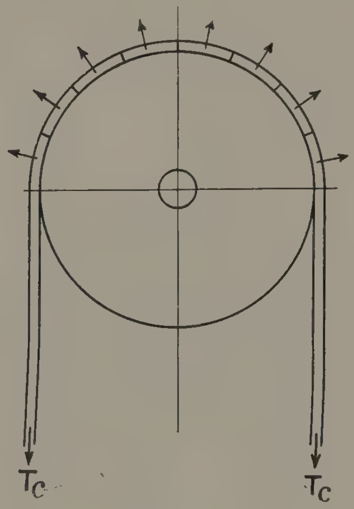


FIG. 7

$$\text{Then } S = \frac{0{\cdot}094 \times 4000 \times 4000}{3600 \times 32{\cdot}2 \times \frac{3}{16}}$$
$$= 69 \text{ lb. per sq. in.}$$

STUDY. What would be the tension per inch width in the above belt. *Ans.* 13·0 lb. Find the stress per square inch and the tension per inch width when V = 4,500, 5,000, 5,500, 6,000. *Ans.* 88, 16·5; 109, 20·2; 132, 24·4; 155, 29·1.

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON X

DRILLING MACHINES WITH MULTIPLE SPINDLES, AND RADIAL MACHINES

The later drilling machines are remarkable designs for accomplishing production in mass without removing the work. Multiple spindle machines have increased amazingly with the extension of motor manufacture. The large numbers of holes of the same, or different sizes to be drilled, reamered, and tapped, handicap the single-spindle machines, which can only be utilized by transporting the work, by hand, if light, between machines; or on tracks when heavy. Hence the growing practice is to build machines with several spindles capable of being set at different centres, and each driven at the rates most suitable. This scheme may take the form of a repetition of the single spindle as in the sensitive machines recently described, but very often the design is different. It is further complicated by the dispositions of spindles, in linear series, in circles, or in rectangular, or irregular patterns, and with provision for individual adjustments, for centres, and for heights. In the last developments for mass products, centres are permanently fixed for repetitive work.

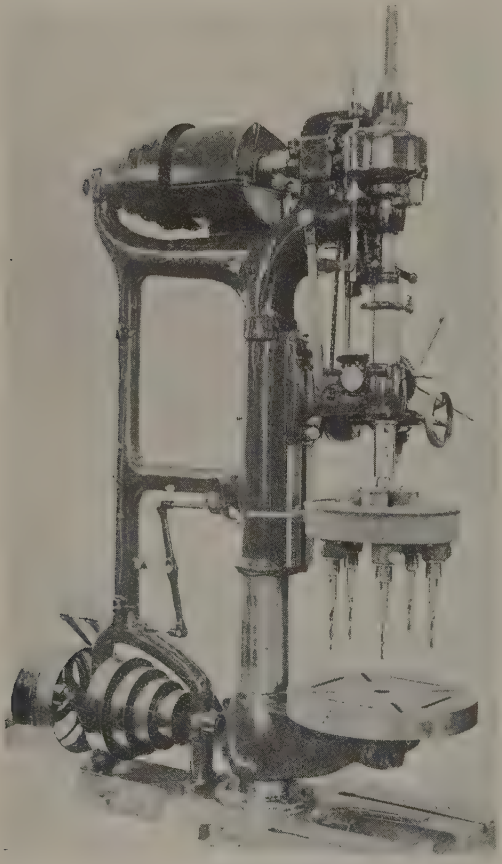


Fig. 51 Upright Drill Carrying a 5 Spindle Head

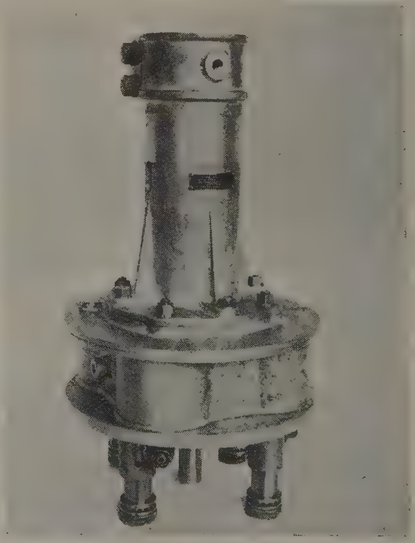


Fig. 52. Head with Circular Disposition of Spindles

Gang Machines. This term includes an immense variety of designs having the common feature of a linear disposition of the drilling spindles. The prototypes of these came into the boiler and plating shops with the advent of steel, where they are still used. The designs that are common may be classed under three heads: those in which the spindles are carried in an attachment mounted on a single machine, those in which the spindle heads are independent, independently driven but supported on a common base, and having a work table in common, and that where the spindles, set at fixed centres, or adjustable, are driven with gears from a horizontal shaft.

An attachment fitted to the spindle of a machine is a concession to the small shop where it is useful for occasional service, and many of these are in use (Fig. 51). A head fits within the taper socket of the spindle, and is clamped to embrace the outside (Fig. 52). The separate spindles are driven from a central gear (Fig. 53), through spur gears, or if they are disposed lineally, with spirals, or in some designs with pitch chain. If the spindles have to be adjustable, then the spiral must be continuous, or if spur gears are used, the drive must be done through idlers, while the pitch chain permits of accommodation. Gears run in oil in an enclosed casing. Heads may carry as many as seven drills, which tax the power of a machine built for one only. An increase in the number of drills means a corresponding reduction in their diameters.

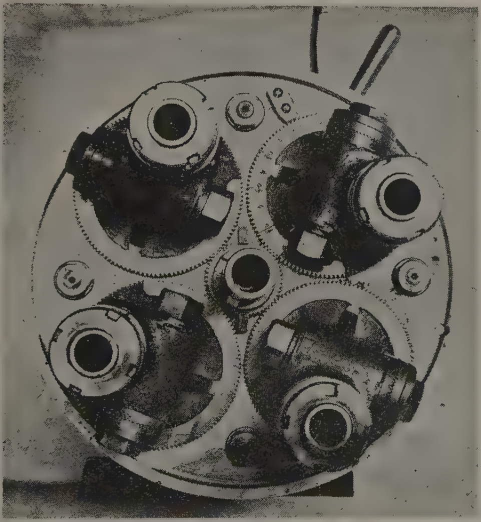


Fig. 53. Underside View of Fig. 52

Machines with independent heads may carry from two to six spindles, driven from a common shaft through gears. They are mounted on a base plate or on a box base with one long table or with separate tables, with provision for vertical settings. Some machines have provision for making lateral adjustments of the heads. The weight of the spindles is counterbalanced. These are driven with belt, or motor, with a large range of geared speeds and feeds, and with drilling capacities up to about $1\frac{3}{4}$ in., high speed.

The machines in which the spindle heads fit to a horizontal rail and are driven in common are a very large group. The features that characterize the machines by the Moline Tool Co. are varied. The heads in any numbers, from six to thirty or more, are carried on a rail and driven with a long spiral, or two spirals of 45° angle. As the spirals are continuous the heads can be set and clamped at any centres. The drive is by belt or motor. There are differences in feeding. In some machines of the heavy types, the feed is imparted to the rail and its heads, the weight of which is counterbalanced. In others, the table, also counterbalanced is fed either with a rack and pinion for regular work, or with cams when the movement is irregular, as when holes in vertical relations are separated with an interspace through which the table is moved at an increased speed. In some cases the table can be traversed longitudinally along its base. Each spindle has its drill lubricated with its separate supply pipe.

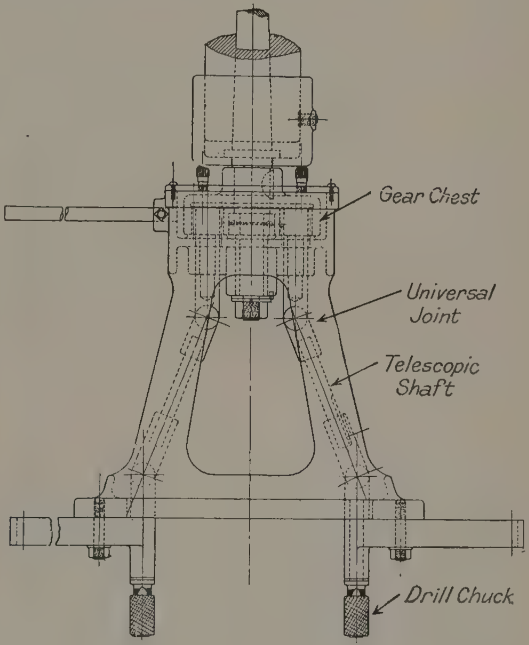


Fig. 54. Portable Multi-spindle Head

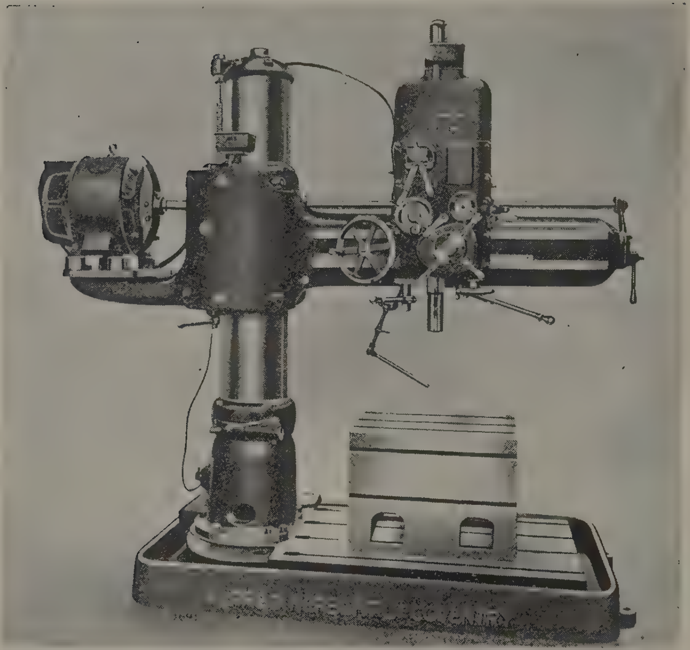


FIG. 55. RADIAL DRILLING MACHINE
(Alfred Herbert Ltd., Coventry)

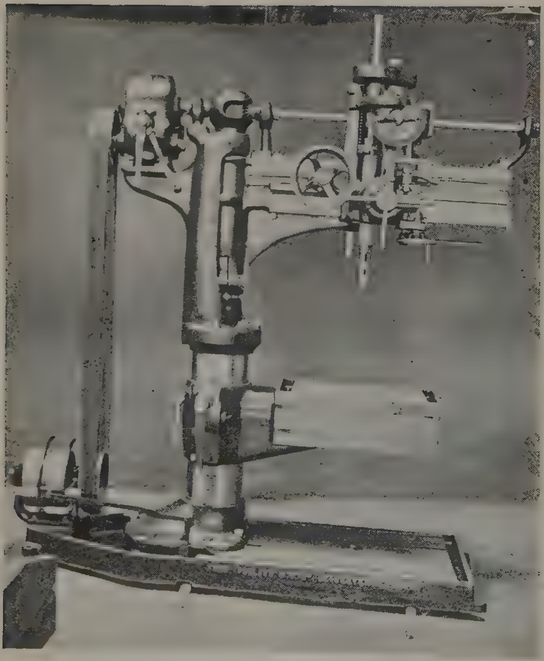


FIG. 56. ALL-GEARED RADIAL MACHINE

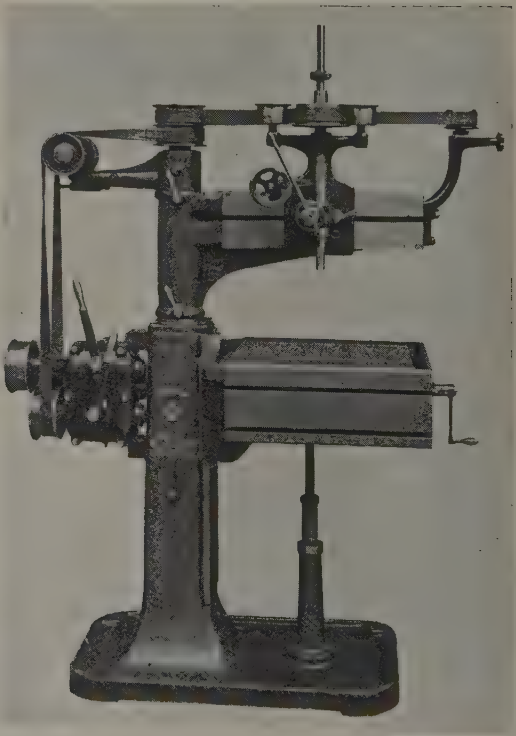


FIG. 57. SENSITIVE RADIAL MACHINE

The Cluster Machines. In these the plan arrangement of the drills is either circular, rectangular, or irregular. The original designs were of German origin, some with vertical, others with horizontal spindles, for drilling all the holes in the flanges of pipes and cylinders at one time; the advantage of the latter design being, that with two heads, the holes in opposite flanges could be dealt with simultaneously. Present machines resemble those, with an extension of utilities and some improvements in details. As in the gang machines, heads carrying several spindles are fitted to single spindle machines (Fig. 54), and they are valuable when the machines are built for heavy duty. Centres may be fixed or adjustable, and the number of drill spindles be varied, but the larger the number the smaller must be the drills carried. For regular service, the machines built for multiple work alone must have preference.

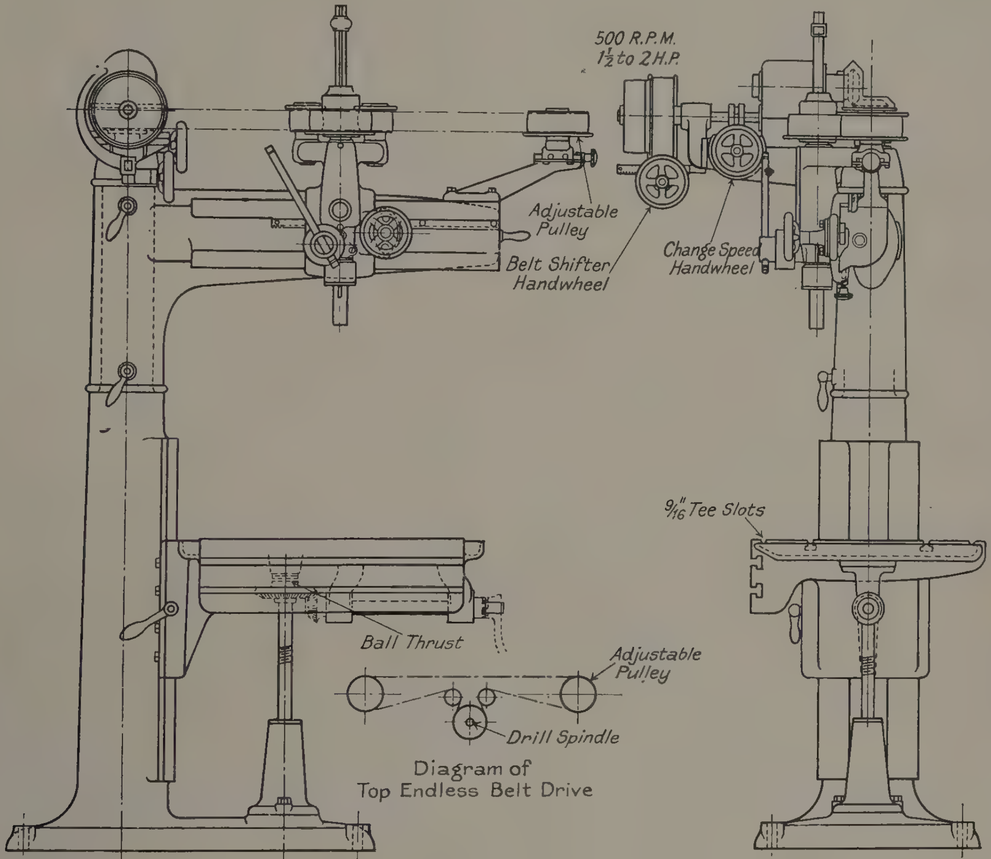


Fig. 58

Designs are varied, but they include a cone-shaped casting, which encloses the spindles and driving gears, and sliding on the face of the column, is counterbalanced. A gear chest in the upper part of the head carries spur gears that transmit motion to the separate drilling spindles, through universal joints, and telescopic shafts, which permit of setting the spindles within a large range of radii. They are clamped in positions with a radius bar that receives the spindle sleeve, against a flange at the bottom of the head. Quick vertical adjustments can be made in the height of individual spindles. Feeding is by hand or power. A standard head is arranged to carry a definite number of spindles, but they need not all be fitted until required. Numbers may range from 8 to 40. As the number of spindles is increased, the sizes of the holes that can be pierced lessens. Thus, a machine will carry 12 1¼ in., 16 1 in., 20 ¾ in., 24 ⅝ in. or 28 ½ in. drills. Generally, the driving gears are permanent, but in several machines provision is made by changing gears for drilling large and small holes at the most appropriate speeds, and for drilling, reaming, spot facing, and counterboring, and spindles not wanted can be thrown out of action.

Radial Arm Machines. All the column machines are handicapped by the limited space between the column and the drill spindle, and by the rigid location of the spindle. The radial arm carrying the spindle head was, therefore, an early design. Except in broad outline and in functions, no resemblance remains between those early machines and their present-day representatives. The early ribbed columns and arms with vertical slides and trunnioned hinging have given place to cylindrical and boxed sections. The overhang of the cantilever arm is limited by the intense pressures of high speed drilling, so that a radius of 6 ft. is nearly the maximum. In the best modern designs the column is cylindrical, turning round a tall central pivot on ball races. This is encircled with a boss that terminates the arm, elevated with a screw. The arm is prolonged at the rear to carry the driving motor (Fig. 55), which also serves as a counterbalance. In a still later design, the column is pierced to receive the arm, also with an extension on which the motor is supported, with the further advantage that the driving shaft goes directly through the column. The arm has vertical adjustments, and it and the column are swung together. The radial arm is now generally of a boxed section, approximately semi-cylindrical, or " pipe " section, on which the spindle saddle is bridged in such a way that the thrust of drilling comes nearly or quite centrally, instead of to one side. And the saddle sometimes runs on rollers to diminish friction.

Of equal value are differences now made in the methods of driving and control. The changes of speeds with belt cones on the column, and of feeds with small cones on the saddle, have been displaced by geared drives for both (Fig. 56), the gears being enclosed in boxes—lubricated, and changed with levers, as in other machine tools. And along with this change the method of centralized control has been adopted, that is, the speed and feed changes are controlled from the saddle, and also the elevation and clamping of the arm, so that the attendant need not move away from his post. The arm is elevated with a screw, which does not always revolve, the rotation being imparted to the nut. The elevation motion cannot be engaged when the arm is clamped, and a safety device prevents the arm from being run beyond the limit of its range. The weight of the spindle is counterbalanced. A large range of spindle speeds and feeds is provided in the larger machines, as many as 16 and 6 respectively. The motor drive is extensively superseding that of a main belt.

The radial machines occur in as large a range as the column types. The heaviest will drill up to 3½ in. in iron and steel, and tap up to 2 in. But there are also very many sensitive machines now built, driven with a high speed belt to the drilling spindle, and maintained in constant tension in all positions of the carriage (Fig. 57). Machines of this class will pierce holes from ¼ in. to 1¼ in. Speeds are changed with a box of gears, they are much higher when ball bearings are used than when they are plain. Thus, with a Jones & Shipman machine (Fig. 58) with plain bearings, the revolutions per minute are: 273, 378, 546, and 788 But with ball bearings they are: 508, 680, 982, and 1,418 r.p.m There are many modifications in the forms of table for lighter and heavier machines, some being made to elevate or to tilt, or to be swung aside. In many machines, spindles will drill at an angle, in many an extra tapping spindle is fitted.

ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

By A. P. YOUNG, O.B.E., M.I.E.E.

LESSON VI

THE EFFECTS OF AN ELECTRIC CURRENT AND THEIR APPLICATIONS

THE presence of a current in an electric circuit is known only by the effects it produces. These manifestations may, broadly speaking, be divided into four distinct classes—

1. The chemical effect.
2. The heating effect
3. The vacuum tube effect.
4. The magnetic effect.

In this lesson, we shall confine our attention to the first three classes only, as the magnetic effect will be dealt with separately under "Electro-Magnetism."

1. THE CHEMICAL EFFECT

Some liquids, like metals, are good conductors of electricity, whilst others are non-conductors. In the latter class we have various grades of oil, which, owing to the extremely high resistance they offer to the passage of electricity, are used extensively as insulators in electrical engineering. Typical examples of the use of oil as an insulator are high-tension power transformers and oil-immersed switchgear.

The liquids, which are good conductors, may be further subdivided into two classes—

(*a*) Liquids which conduct without decomposition, examples of which are liquid mercury and molten metals.

(*b*) Liquids known as "Electrolytes," in which the passage of a current is accompanied by decomposition of the liquid into its constituent components. In this class, we have water, copper sulphate, silver nitrate, etc.

In considering the chemical effect of a current, we are concerned with the manifestations of an electric current in liquids of the electrolyte class. In Fig. 23 a simple electrolytic cell is illustrated, and there we see that the current enters the cell by way of the "Anode," passes through the "Electrolyte" to the "Kathode," from which plate it leaves the cell.

These terms were introduced by Michael Faraday in describing his classical experiments on this subject. He also referred to the decomposition of a liquid by an electric current, as "Electrolysis," and assigned the name "ions" to denote the components or elements liberated as a result of electrolysis.

As examples of electrolytes, we may mention water and copper sulphate. Perfectly pure water is practically a non-conductor, but a trace of acid is sufficient to make it a relatively good conductor. The passage of a current through water will result in its decomposition in accordance with the equation

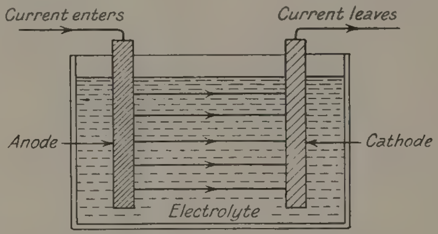


FIG. 23. SIMPLE ELECTROLYTIC CELL

$$2\,H_2O = 2\,H_2 + O_2$$

The volume of hydrogen liberated at the kathode will be twice that of the oxygen liberated at the anode. If the two gases be collected together, they can be combined again to form a drop of water, if ignited by an electric spark.

Copper sulphate also is readily decomposed by electrolysis, the metallic copper in the solution being liberated and deposited at the kathode. The chemical changes which take place are represented by the equations—

$$Cu\,SO_4 = Cu + SO_4 \quad . \quad . \quad . \quad . \quad (12)$$

(Copper Sulphate) = (Copper) + (Sulphion)

and,

$$SO_4 + H_2O = H_2\,SO_4 + O \quad . \quad . \quad (13)$$

(Sulphion) + (Water) = (Sulphuric acid) + (Oxygen)

The combination represented by equation (13) only occurs if platinum electrodes are used, the oxygen (O) appearing at the anode. When copper electrodes are employed (as in electroplating) no oxygen appears, as the copper anode is dissolved and taken into solution at the same rate as copper is deposited at the kathode.

Faraday's Laws of Electrolysis. The laws governing electrolysis were deduced by Faraday and may be briefly stated as follows—

(*a*) The amount of chemical action is the same at all points of the circuit.

(*b*) The weight of the ions liberated at an electrode in a given time is proportional to the strength of the current.

(*c*) The weight of the ions liberated at an electrode in one second is equal to the strength of the current multiplied by the electro-chemical equivalent of the ion.

These three laws may be expressed in a very simple formula, from which the weight of a metal or gas liberated in an electrolytic cell can be calculated for given conditions.

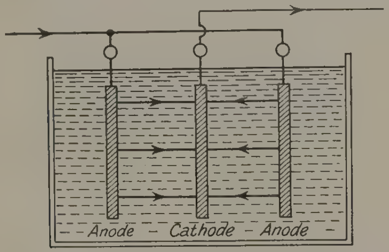


FIG. 24. CELL FOR ELECTRO-PLATING

$$W = z \times I \times t \quad . \quad . \quad . \quad . \quad (14)$$

Where,

W = weight of elements (in grm.) liberated
I = current in amperes
t = time in seconds, and
z = electro-chemical equivalent (E.C.E.).

If Q = quantity of electricity in coulombs, then

$$W = z \times Q \quad . \quad . \quad . \quad . \quad (15)$$

Use is made of equation (15) in the design of different types of electrolytic meters, known as quantity or ampere-hour meters, for measuring the quantity of electricity passing in a circuit. If it is assumed that the voltage remains constant, the quantity of electricity passing in a given time will be directly proportional to the energy, since

$$\text{Energy} = Q \times E$$

As in all electric supply systems, the voltage is maintained practically constant, such meters can be and are calibrated to read electrical energy in "Units."

The electro-chemical equivalent (z) of an element is the weight in grammes liberated by the passage of one coulomb of electricity, and is directly proportional to the chemical equivalent. It is, in fact, the chemical equivalent multiplied by 0·00001044. The values of z for a few of the more important elements, are given below—

Element	z (*Grammes per Coulomb*)
Hydrogen	0·00001044
Copper (Cupric)	0·00032945
Copper (Cuprous)	0·00065782
Silver	0·0011183
Oxygen	0·00008292
Nickel	0·00030411
Gold	0·00068129
Zinc	0·00033902
Potassium	0·00040783

Referring to equations (14) and (15) these may be rewritten as—

$$I = \frac{W}{zt} \quad \text{and} \quad Q = \frac{W}{z}$$

and, from these equations, it is obvious that an electrolytic cell can be utilized to measure accurately either the strength of a current or the quantity of electricity in a circuit. When an electrolytic cell is used for this purpose it is called a "Voltameter."

In applying the principles of electrolysis to electro-plating, the object to be plated is made the kathode, and is usually placed between two anode electrodes as shown in Fig. 24.

The solutions generally used are—

Copper plating	Copper sulphate.
Nickel plating	Double sulphate of nickel and ammonium.
Gold plating	Double cyanide of gold and potassium.
Silver plating	Double cyanide of silver and potassium.

In addition to the industrial applications already mentioned, the principles of electrolysis are applied in electrolytic furnaces for the reduction of metals from their ores. For instance, aluminium is produced in this way from the mineral bauxite; also, pure copper (known as electrolytic copper) as used for electrical conductors, is refined by an electrolytic process.

2. HEATING EFFECT

It was explained in Lesson III, that when an electric current flows through a conductor, heat is generated, i.e. a portion of the electrical energy is converted into heat energy. Whilst in electrical machinery this is very undesirable, since it results in inefficiency, in other types of apparatus, such as electric heating and cooking appliances, it is necessary that the maximum amount of heat energy be developed.

Joule found in investigating the relation between energy expended and heat generated, that 4·2 joules of work are necessary in order to raise 1 grm. of water 1° C., which corresponds to the French unit of heat called the "Calorie." Thus, we have a link between electrical energy and heat energy.

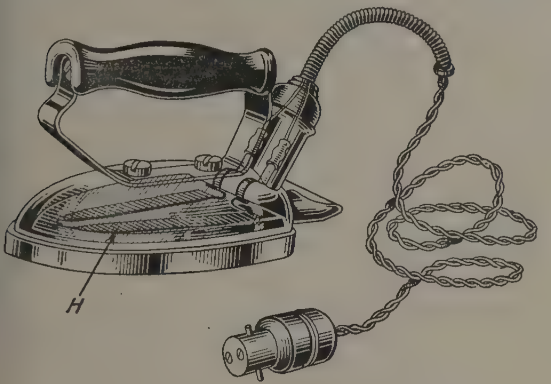


FIG. 25. AN ELECTRIC FLAT IRON

H = Heating element. High resistance alloy wire wound round strips of sheet mica. Sheets of mica are placed above and below the heating element for insulation purposes.

The energy expended in an electric circuit is expressed by the equation—

Energy $= I.E.t$ joules, where
I = current in amperes,
E = E.M.F. in volts, and
t = time in seconds.

Since, $E = I.R.$, the equation may be written—

Energy $= I^2 R.t$ joules
Where, R = resistance in ohms.

If no mechanical work is done, then all the electrical energy will be converted into heat energy, and from the relationship between heat and electrical energy, obtained by Joule, the following equation, known as "Joule's Law," is derived—

$$U = \frac{I^2 R.t}{4{\cdot}2} \text{ calories}$$

$$U = 0{\cdot}24\ I^2 Rt \text{ calories} \quad . \quad . \quad . \quad (16)$$

From the foregoing, we can say that the number of heat units generated in any conductor is proportional to—

(a) The resistance of the conductor,
(b) The square of the current, and
(c) The time during which the current flows.

The following example will illustrate the practical application of equation (16).

EXAMPLE. A 500-watt electric kettle takes 8 minutes to boil 1 pint of water. What is the efficiency of the kettle if the initial temperature of the water was 15° C.?

Heat energy developed electrically

$$= U_1 = 0{\cdot}24\ I^2\ Rt \text{ calories}$$

Now,

$$I^2R = IR \times I = E \times I = \text{watts} = W$$
$$U_1 = 0{\cdot}24\ W.t \text{ calories,}$$
$$U_1 = 0{\cdot}24 \times 500 \times 8 \times 60$$
$$= 57{,}600 \text{ calories.}$$

Heat energy communicated to the water

$$= U_2 = \text{mass (grm.)} \times \text{temp. rise (°C.),}$$
$$1 \text{ pt. of water} = 1{\cdot}25 \text{ lb.} = 1{\cdot}25 \times 453{\cdot}6 \text{ grm.}$$
$$U_2 = 1{\cdot}25 \times 453{\cdot}6 \times (100 - 15)$$
$$= 48{,}200 \text{ calories.}$$

$$\text{Efficiency of kettle} = \frac{48{,}200}{57{,}600} \times 100$$
$$= 83{\cdot}6 \text{ per cent.}$$

The heating effect of a current finds practical application in numerous ways, the more important being—

(a) Electric measuring instruments, known as "Hot Wire" instruments.
(b) Electric filament lamps.

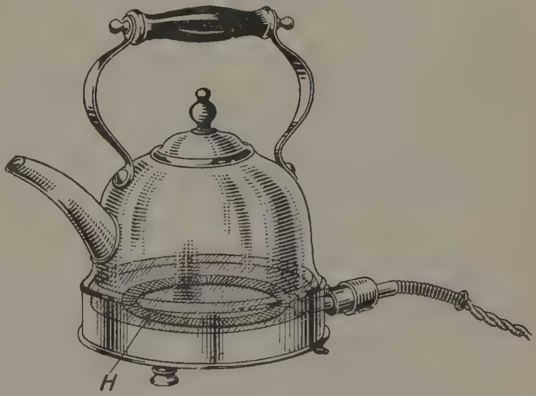


FIG. 26. AN ELECTRIC KETTLE

H = Heating unit. High resistance alloy wire wound in sheet mica the heating unit is insulated from the kettle by mica plates placed above and below the unit

(c) Electrical domestic appliances, such as radiators, kettles, flat irons, toasters, etc. Figs. 25 and 26 show the arrangement of heating units in electric flat irons and kettles.
(d) Electric fuses.

(*e*) Surgical appliances.

(*f*) Electric furnaces which utilize the intense heat of the electric arc.

(*g*) Electric welding in which the two metals to be welded are held together under pressure, and current passed through the surface of contact. This method is known as spot welding, and was first introduced by Prof. Elihu Thomson, in America.

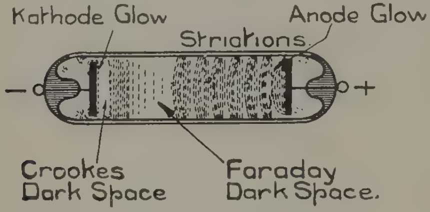


Fig 27. Vacuum Tube

3. The Vacuum Tube Effect of a Current

The electric discharge in a partial vacuum (see Fig. 27) is indeed a pretty sight. The kathode exhibits a bluish or violet glow which is separated from the electrode by a dark space. At the anode there is a small bright star of light. The discharge between the electrodes breaks up into a series of striae, with a certain degree of exhaustion, and these vibrate to and fro. The light emitted by the discharge is rich in those rays which produce phosphorescence and fluorescence.

Sir W. Crookes found that if the tube be practically exhausted (so that only about one-millionth of the air remains) the whole of the tube becomes dark and only the walls appear phosphorescent. The discharge is then purely electronic, and any object (such as a cross) placed between the electrodes will cast a shadow. The electronic stream, if allowed to impinge on a metal target, will produce the wonderful X-ray. This was discovered accidentally by Röntgen in 1895, when experimenting with a Crookes tube.

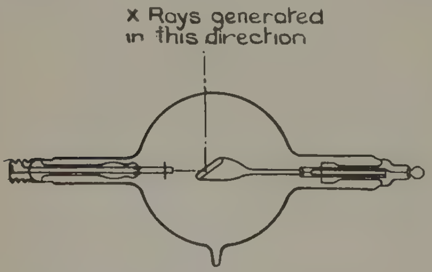


Fig. 28. Coolidge X-ray Tube

In Fig. 28 is shown a Coolidge X-ray tube, the kathode of which was invented by Dr. Coolidge of the General Electric Company of America. The kathode is a spiral of tungsten wire maintained at incandescence by an auxiliary source of current, whilst the anode is pure tungsten or tungsten let into copper. The discharge consists of a pure electronic stream from kathode to anode, and in impinging on the tungsten target at high velocity, generates X-rays and, incidentally, a very large amount of heat. Large copper fins are, therefore, provided on the end of the anode to facilitate cooling. The applications of X-rays in medical science are too well known to need explanation here. It is not, however, so generally known that this effect is largely employed in the engineering industries, one important application being the detection of flaws in castings.

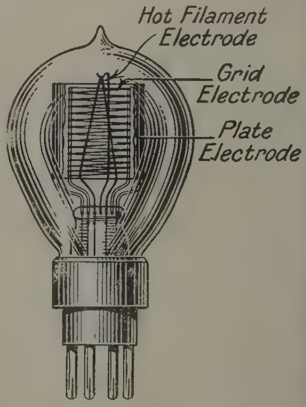


Fig. 29. A Radio Power Amplifying Valve

The tremendous advance in electrical science during the past two decades, resulting in the new "Electron Theory," is due solely to a careful investigation of electrical phenomena in a nearly perfect vacuum. By peering into nothingness, with spectacles fashioned by science, man has revealed a vision of greatness. Wireless broadcasting—or radio, as it is called—which is having a profound effect on the future course of civilization, has become a reality through the evolution of the modern "valve" comprising two electrodes (one a hot filament) as used in the early Fleming valve, with a third "grid" electrode, introduced by De Forest. By means of the grid, the electronic stream between the hot filament and the plate can be controlled, and the incoming signals, which vary the potential or pressure on the grid, are thereby amplified.

Fig. 29 illustrates a power amplifying valve of the dull emitter type.

JIGS AND TOOLS

By Leo Kelsey, A.M.I.P.E.

LESSON VI

MILLING CUTTERS AND FIXTURES

THE cutting tools used on milling machines have wo characteristics. They all rotate and all have venly spaced teeth. In Figs. 42–48 the standard forms are shown, together with their usual names.

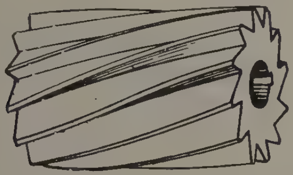


Fig. 42. Spiral Roller Mill

Form or contour milling cutters have a cer-ain form of tooth construction which ensures he same contour or profile for the cutter, however many times it may be resharpened. This s achieved by what is known as “ backing-off.”

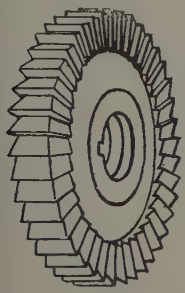


Fig. 43. Side and Face Mill
(If used in Pairs “ Straddle ” Mills)

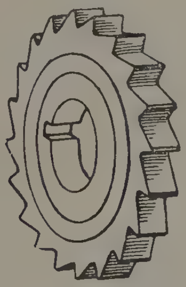


Fig. 44. Slotting Cutter

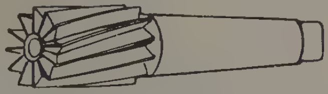


Fig. 45. Spiral End Mill

Backed-off cutters have teeth which are of the ame cross-section throughout their depth ; this s shown in Fig. 49, where the difference from an ordinary cutter is shown. Milling cutters for gears are made in this manner. Owing to the introduction of special spur and bevel gear cutting machinery, milling machines are no longer used in modern plants for gear cutting. They are necessarily used to a large extent in tool-rooms and small plants, where the gear cutting work is not of sufficient quantity to warrant a special machine.

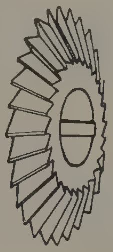


Fig. 46. Equal Angle Cutter

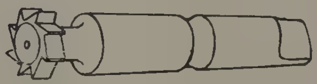


Fig. 47. Tee Slot Cutter

Inserted tooth milling cutters are becoming increasingly popular on account of their economy. The method of constructing these cutters is shown in Fig. 50. The body *A* is of mild steel, and is made solid or detachable from the shank or arbor. The pins *H* are tapered and expand the body when driven home, thus gripping the tool steel cutting bits *P*. When the cutter is wearing down owing to resharpening, all that is necessary to return

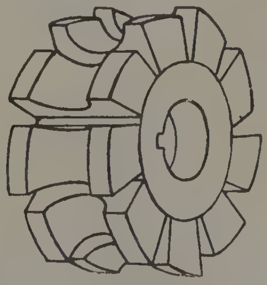


Fig. 48. Backed-off Form Mill

it to its original form, is to slacken the pins *H* and push the bits *P* down their slots, drive home the pins again and sharpen by grinding. There are numerous methods of holding the inserted bits in the cutter body, but they are all adjustable, and only differ from the simple method of a taper pin, in that more elaborate precautions are taken to keep the cutters in position while operating.

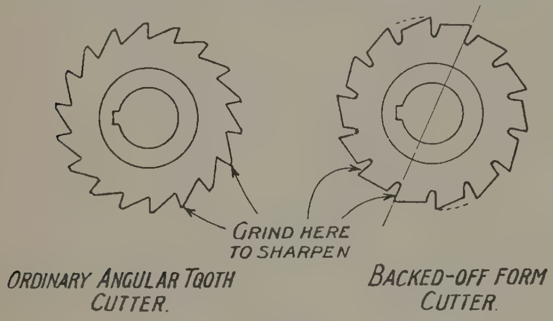


FIG. 49

Milling Fixture Design is in some respects somewhat easier than drilling jigs, because more space is allowed and weight, instead of being a disadvantage, becomes the opposite. Metal should never be spared in milling fixture design, as the intermittent motion of the cutters produces vibration, which, unless absorbed in the fixture, will produce work with uneven surfaces, having tool or "chatter" marks on it. Referring to Fig. 34, in the article on clamping, the milling fixture there is of typical design, and conforms to our previous statements on the functions of jigs. The clamping is simple and rapid, the fixture is easy to load, the locating surfaces being visible and easy to clean. The group milling cutters or "gang" cutters, show a typical method of finishing to size at one pass of the cutter a very uneven contour. The cutting load on the component in this case is very heavy, and only components of a substantial nature should be cut in this manner.

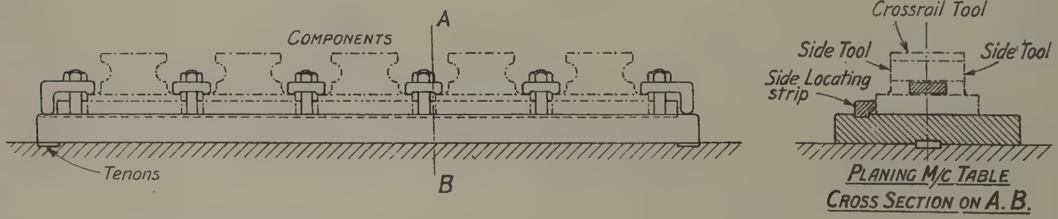


FIG. 51

An improvement, which would not be at all difficult in the fixture just described, would be to extend the length so that, instead of four components being held, twelve could be clamped in position. The advantage of this is that when the last component on the fixture is passing under the cutter, all those which have preceded it are clear of the latter and can, therefore, be removed from the fixture, and the locating faces cleaned preparatory to reloading.

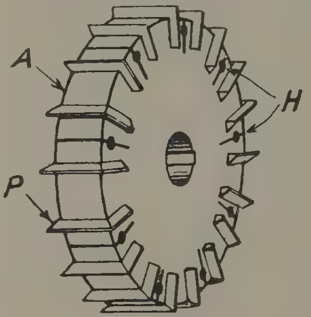


FIG. 50. INSERTED TOOTH CUTTER

Detachable Vice Jaws have already been referred to; they must not be disregarded, however, for whenever accuracy is not the first essential, and quantities are not large, they often represent the cheapest and most economical of milling fixtures; they are cheap in first cost, have long life, and hold the component rigidly while it is being cut. Much ingenuity can be spent on their design and construction, and it is well repaid.

Planing Machines do not lend themselves readily to the rapid production of components which are not of fairly large dimensions and heavy structure. However, by suitably arranging the components, "gang" planing is very economical. This form of planing is much the same as "gang" milling, except, of course, that the tool is of single point and not "formed" like the milling cutters. A careful study of the ang planing fixture in Fig. 51 will show that ery little provision is made for lateral dislacement of the components, the reasons for his were explained in the paragraph in planer nd shaper tool action. Four tools are used in laning the component shown, two on the crossail and two on the side-heads. Planing fixtures re made of massive proportions, as although veight is an objection in that it increases the nertia load on the table reversing mechanism, t must provide sufficient metal to absorb the shock of the interruption of the tool finishing the cut on one component and beginning to cut another. Rigid clamps are necessary with large nuts and bolts.

The Shaping Machine is now so little used for production in quantities that it is not necessary to go into fixture design for these machines. However, the shaping machine is still largely used in general engineering and also in the toolroom, where the universal shaping machine is an exceedingly useful tool.

ENGINEERING MATERIALS

TESTING AND PROPERTIES

By R. G. Batson, A.K.C., M.Inst.C.E., M.I.Mech.E.

LESSON V

MEASUREMENTS OF ELASTICITY

THE total elongation of a test piece after fracure is usually measured to an accuracy of about ± 2 per cent. In order to obtain the elastic xtension at the limit of proportionality with he same percentage accuracy it is necessary to neasure that extension correctly to 0·00004 in. t is difficult to appreciate a length of this value, vhich is about one-fortieth of the thickness of a nodern cigarette paper.

A modern extensometer measures to ·000008 in., while a lateral strain meter reads o 0·000004 in. It has been found that the alue of the limit of proportionality of some naterials depends to a large extent on the ccuracy of the measuring instrument employed, wing to the fact that the deviation from proporionality is small and gradual.

EXTENSOMETERS

The principal instruments in use in this ountry for the measurement of longitudinal trains employ the following general methods—

1. A single micrometer screw combined with multiplying lever, e.g. the Cambridge extensoneter.
2. A microscope and multiplying lever, e.g. he Ewing extensometer.
3. Optical magnification, e.g. Lamb's extenometer, or Martens mirror extensometer.

The Cambridge Extensometer. This instrument made by the Cambridge Instrument Co., Ltd., Cambridge, is shown diagrammatically in Fig. 18. It consists of two frames *A* and *B*, each attached to the test piece *T* by two knurled-headed pointed screws at *C* and *D*. The lower frame carries a micrometer screw *E*, with a hardened point *H*, and a vertical arm *F*. The upper frame rests on a knife-edge *G* on this vertical arm, and is continued by a flexible tongue *K*. This combination forms a 5 to 1 lever.

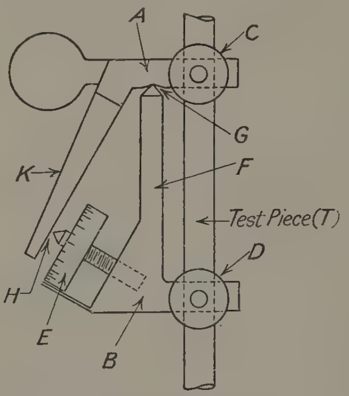


FIG. 18. THE CAMBRIDGE EXTENSOMETER

Extension of the test piece lowers the tongue, the tongue is vibrated, and the movement of the micrometer screw to touch the tongue is, therefore, a measure of extension on the gauge length CD. The instrument is made for gauge lengths of 2 and 4 in., and readings can be repeated to 0·00004 in.

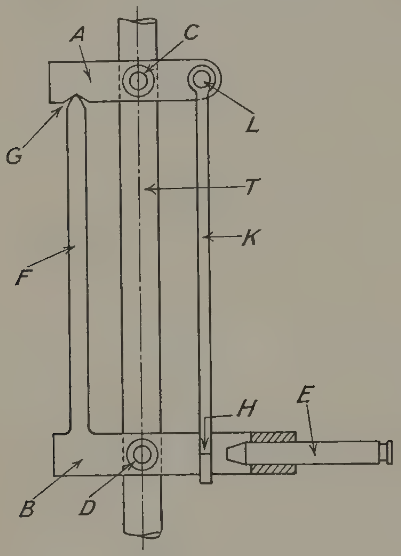


FIG. 19. THE EWING EXTENSOMETER

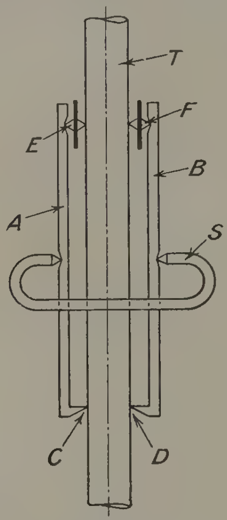


FIG. 20. MIRROR EXTENSOMETER

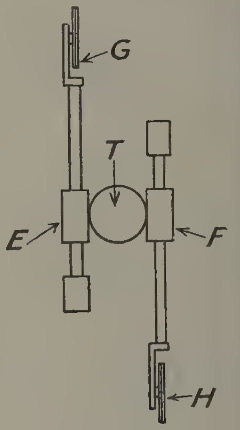


FIG. 21. MIRRORS AND DOUBLE KNIFE EDGES OF MIRROR EXTENSOMETER

The Ewing Extensometer. This instrument, also made by the Cambridge Instrument Co., Ltd., is used in a large number of testing laboratories. It is shown diagrammatically in Fig. 19. The two frames A and B are attached to the test piece T by pointed screws at C and D. The lower frame carries (1) a vertical post F, terminating in a ball point G, and (2) a microscope E. The upper frame has a conical recess which rests on the ball point and also carries, on the opposite side of the test piece, a sighting rod K suspended from L. Extension of the test piece causes the upper frame to tilt about the point G as a fulcrum. The sighting rod K is raised, and the movement of the cross wire H on the rod is measured by a scale in the eye piece of the microscope. Scale readings can be estimated to 0·00002 in., and the instrument is made for gauge lengths of 2 and 8 in., and 10 and 20 cm.

Martens Mirror Extensometer. An instrument of this type is made by Alfred J. Amsler & Co. It consists of two parts—

1. The mirror apparatus.
2. The telescope observation apparatus.

The mirror apparatus is shown diagrammatically in Fig. 20. Two clips A and B are held on opposite sides of the test piece T by a spring S. Each clip has a knife-edge C and D at one end which is held directly on the test piece. The other end has a double knife-edge E and F interposed between it and the test piece. Each double knife-edge is attached to a rod and mirror G and H, as shown in Fig. 21. Extension of the test piece causes rotation of the double knife-edge, and therefore of the mirrors. The amount of this rotation is measured by a telescope observation apparatus, the principle of which is as follows (see Fig. 22)—

a = diameter of the double knife-edge.
b = scale distance.
c = scale reading.

If the test piece extends a length x, then the mirror turns through an angle α, and

$$a \sin \alpha = x \quad . \quad . \quad . \quad . \quad . \quad (1)$$

According to the laws of light, $c = b \tan 2\alpha$

If the angle α is small, $\tan 2\alpha = 2 \tan \alpha = 2 \sin \alpha$ and $c = b \tan 2\alpha = 2b \sin \alpha$

$$\text{or } \sin \alpha = \frac{c}{2b}$$

$$\text{and from (1) } x = a \sin \alpha = \frac{ac}{2b}$$

$$\text{or } c = \frac{2bx}{a}$$

The value of the extension on a gauge length $= d$ can be measured to 0·000008 in.

Determination of the Modulus of Direct Elasticity

The diameter of the test piece is measured, and the gauge points marked on it. It is then "set up" in the testing machine, the extensometer placed in position, and all the necessary adjustments carried out.

Extensometer readings are taken with gradually increasing loads; the increments of load should be about one-fifteenth of the estimated value of the limit of proportionality. The observations are plotted as a load-extension diagram, or the corresponding stresses and strains can be calculated and a stress-strain diagram drawn.

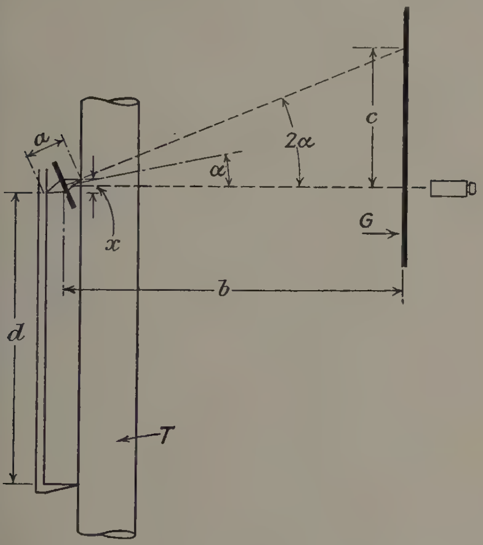


Fig. 22

A typical set of observations for "Armco" iron are given in Table II, and plotted in Fig. 23.

With this test the diameter of the test piece was 0·701 in., and the cross-sectional area was 0·386 sq. in. The full gauge length was 5 in., but the extensions by the extensometer were measured on a gauge length of 3 in.

From the curve (Fig. 23) it will be seen that the limit of proportionality was 9·6 tons per sq. in., and the yield stress was 14·25 tons per sq. in.

The strain corresponding to a stress of 9 tons per sq. in. was 0·000680.

If A = cross-sectional area of the test piece in sq. in.
L = gauge length in inches.
P = load in tons.
x = extension in inches at a load of P tons.

Then E (modulus of elasticity) $= \dfrac{\text{stress}}{\text{strain}}$

$$= \frac{P}{A} \div \frac{x}{L} = \frac{PL}{Ax} \text{ tons per sq. in.}$$

In the above example

$$E = \frac{\text{stress}}{\text{strain}} = \frac{9 \times 2240}{0{\cdot}000680} = 29{\cdot}7 \times 10^6 \text{ lb per sq. in.}$$

TABLE II

Load in tons	Stress in tons per sq. in.	Extension on 3 in. in inches	Strain
0	0	0	0
0·25	0·65	0·000159	0·000053
0·50	1·30	0·000294	0·000098
0·75	1·94	0·000438	0·000146
1·00	2·59	0·000585	0·000195
1·25	3·24	0·000729	0·000243
1·50	3·89	0·000876	0·000292
1·75	4·54	0·001014	0·000338
2·00	5·18	0·001170	0·000390
2·25	5·83	0·001311	0·000437
2·50	6·48	0·001458	0·000486
2·75	7·13	0·001620	0·000540
3·00	7·78	0·001761	0·000587
3·25	8·42	0·001908	0·000636
3·50	9·07	0·002058	0·000686
3·75	9·72	0·002208	0·000736
4·00	10·37	0·002358	0·000786
4·25	11·01	0·002514	0·000838
4·50	11·66	0·002661	0·000887
4·75	12·31	0·002826	0·000942
5·00	12·96	0·002988	0·000996
5·25	13·61	0·003189	0·001063
5·50	14·25	Yield	Yield

Lateral Strain Meters

The longitudinal extension of a bar when under load is accompanied by a transverse or lateral contraction. The ratio of the lateral strain to the longitudinal strain is called *Poisson's Ratio*, and is usually indicated by $\frac{1}{m}$.

Although this ratio is generally obtained indirectly by means of two tests, viz., a torsion test and a tensile test, the lateral strain of a test piece can be obtained at the same time as the longitudinal strain is measured.

A strain meter for this purpose has been designed by Prof. E. H. Lamb, of East London College. It is made commercially by A. Macklow Smith, 15 and 16 Queen Anne's Chambers, Westminster.

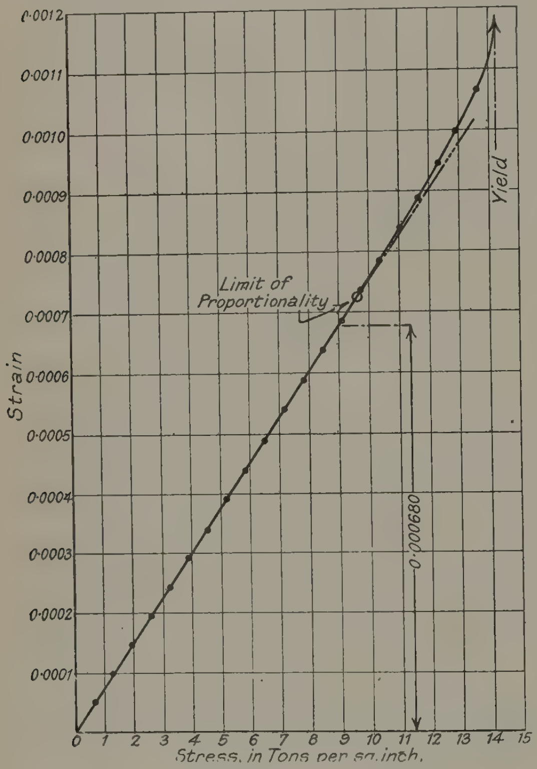


FIG. 23. STRESS-STRAIN CURVE

The instrument is shown diagrammatically in Fig. 24. It consists of two rigid cross-members *A* and *B* held together on either side of the test piece by two parallel bars *E* and *F*. One member *A* has an adjustable stop *G*, which rests on one side of the test piece. The other member *B* carries a moving block *H*, held against the other side of the test piece by spring pressure. The moving block is guided by two small measuring rollers *C* and *D*, and by two steel balls running in vee grooves. Each of the measuring rollers carries a small mirror as indicated in the diagram.

A reduction in the diameter of the test piece causes a movement of the block *H*, accompanied by a rotation of the measuring rollers, and therefore of the mirrors. This rotation is measured by a telescope as indicated.

If y = the reduction of diameter of the test piece in inches.
x = the scale reading in centimetres.
d = the mean diameter of the rollers C and D in inches.
L = the distance from the scale to the mirror facing it, in centimetres.
a = distance between the mirrors in centimetres.

then the optical arrangement is such that

$$y = \left(\frac{d}{4L + 2a}\right) x$$

Then the lateral strain is "y" divided by the diameter of the test piece in inches.

A series of observations are taken with increasing loads, and the readings plotted in the same way as for the determination of the modulus of direct elasticity.

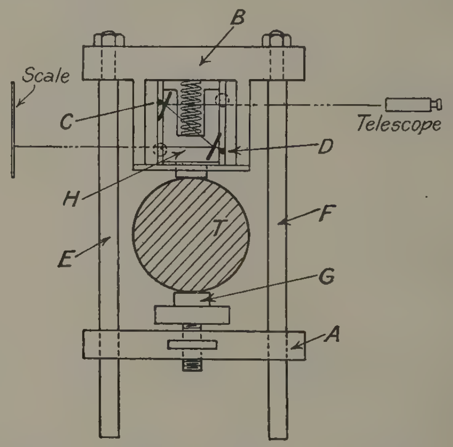


FIG. 24. LAMB'S LATERAL STRAIN METER

The Martens mirror or similar extensometer can be used on the test piece at the same time as the lateral strain meter.

If D = the diameter of the test bar,
P = the load in tons,
x = the extension in inches on a gauge length of L inches,
y = the reduction in diameter in inches,

then longitudinal strain $= \dfrac{x}{L}$

lateral strain $= \dfrac{y}{D}$

$$\text{Poisson's ratio} = \frac{\text{lateral strain}}{\text{longitudinal strain}} = \frac{y}{D} \div \frac{x}{L}$$

$$= \frac{yL}{xD}$$

For the bar of "Armco" iron previously described, the lateral strain for a stress of 9 tons per sq. in. was 0·000195. As the longitudinal strain for this stress was 0·000680

$$\text{Poisson's ratio} = \frac{0{\cdot}000195}{0{\cdot}000680} = 0{\cdot}29$$

The value for steel varies from 0·27 to 0·30. For copper the value is slightly higher (0·31 to 0·34).

Prof. Lamb has designed an extensometer on the same roller principle that he has adopted for his lateral strain meter.

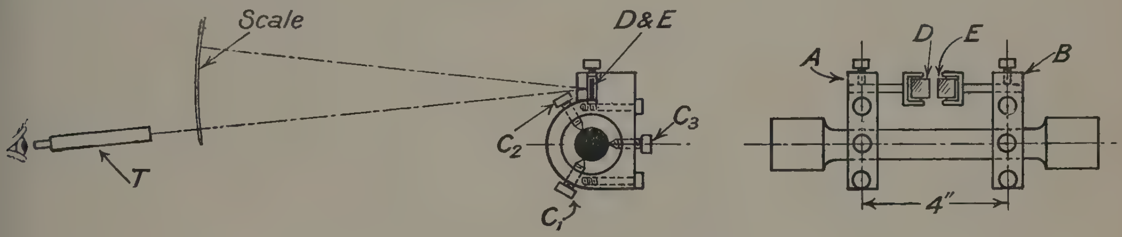


FIG. 25. MIRROR TORQUEMETER

THE MODULUS OF RIGIDITY

Tangential loading or shear produces shear strain and shear stress.

Within the elastic limit, shear stress divided by the accompanying shear strain is a constant which is called the *modulus of transverse elasticity* or *modulus of rigidity*, and is usually denoted by C or N.

There is a mathematical connection between the modulus of direct elasticity E, the modulus of rigidity N, and Poisson's ratio $\left(\frac{1}{m}\right)$

$$\frac{1}{m} = \frac{E - 2N}{2N}$$

The easiest way of determining the value of N is by means of a torsion test. Torsion is the shear that is produced when one layer of a body is made to rotate on the next.

If D = the diameter of the test piece,
L = the gauge length,
θ = angle of twist of the test piece in degrees on a gauge length of L, and under a twisting moment of T.

$$\text{then } N = 583 \frac{TL}{D^4\theta}$$

The relation between T and θ is found by placing a test piece under an increasing value of the twisting moment T applied by a torsion testing machine, and measuring the corresponding values of θ by means of a torque meter. Various types of torque meters have been designed from time to time. A simple type of mirror instrument is shown diagrammatically in Fig. 25. In this instrument two small clips A and B are fastened to the test piece by pointed screws C_1, C_2, and C_3, on a gauge length of 4 in. Mirrors D and E of a Martens mirror extensometer are attached to the clips. The mirrors are arranged on a vertical plane through the axis of the test piece, and their movements are measured by means of a telescope T. The difference between the movements of the two mirrors is a measure of the angle of twist on a gauge length of 4 in.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

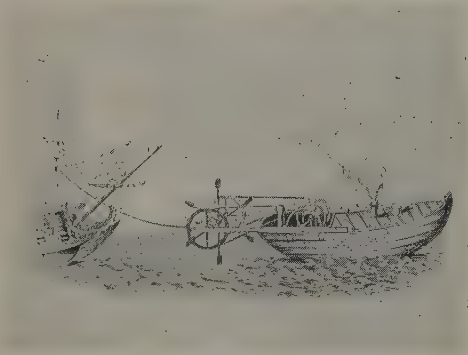
LESSON XI

THE COMING OF THE STEAMSHIP

JONATHAN HULLS

Jonathan Hull,
With his paper skull,
Tried hard to make a machine
That should go against wind and tide:
But he, like an ass,
Couldn't bring it to pass,
So at last was ashamed to be seen.

So ran a doggerel verse for many years after Jonathan Hulls, one of the earliest pioneers of the steamship, had passed away. A prophet has, indeed, very little honour in his own country, and Hulls, on account of his endeavours to make a steamboat "that should go against wind and tide" found himself subjected to so much ridicule, that he eventually gave up his experiments in disgust.



Jonathan Hulls' Steamboat

Very little is known of the history of Jonathan Hulls, with the exception of the fact that he was of Gloucestershire origin, being born about the year 1699 in the village of Campden. Very early in his life, Hulls began to make experiments with model steamboats, and in 1737 he published a treatise, the outcome of these experiments, entitled: *A Description and Draught of a new-invented Machine for carrying Vessels or Ships out of or into any Harbour, Port, or River, against Wind or Tide, and in a calm.*

In brief, Hulls' idea was to place a Newcomen atmospheric steam-engine in a stoutly built boat or barge, and to cause it to operate a paddle wheel placed astern. Hulls proposed to obtain the necessary rotary motion from the Newcomen engine by means of a series of ratchets, weights, and ropes. The fundamental idea of the crankshaft escaped him, however, and it was on account of the cumbersome method of obtaining rotary motion from his engine, besides many inherent defects in the engine itself, that Hulls' steamboat proved to be a practical failure.

Hulls made experiments with his model boats on the River Avon, near Evesham. His great experiment and practical trial took place on the Avon at Evesham in 1737, a considerable number of spectators being present. The trial was an entire failure, and, instead of being given due credit for his endeavours, Hulls, as we have seen, came in merely for a torrent of ridicule.

So passed away Hulls' attempts at steamboat construction. After the year 1737, Hulls' steamboat was never given a further trial, and its inventor relapsed into comparative obscurity, In 1754, he published a treatise on mensuration. and afterwards a volume on the art of brewing. But he had finished with the steamboat, and further years had still to elapse before other minds took up the subject and made a practical success of it.

Such, in brief, is the curious history of Britain's first steamboat projector.

THE SLIDE-RULE

By W. G. Bickley, M.Sc.
Lecturer in Mathematics, Battersea Polytechnic

LESSON II

Use of Slide-rule. Having explained the principles and construction of the ordinary straight slide-rule, we proceed to give some hints as to how it can be most efficiently used. Any one setting of the rule will enable us to obtain a set of pairs of numbers in simple proportion; Fig. 2 illustrates this, for there, for any pair of opposite graduations, c on the **C** scale and d on the **D** scale, we have

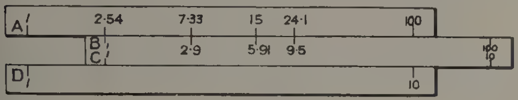


Fig. 7

$$\frac{c}{d} = \frac{1}{2}$$

Example 1. To convert inches to centimetres. Since 1 in. = 2·54 cm., we have to multiply the number of inches by 2·54 to obtain the equivalent number of centimetres. We must therefore set the 1 on one scale against the 2·54 on the other. Using the **C** and **D** scales, some of the **C** scale will protrude to the right. It is often more convenient to use the **A** and **B** scales. Set the 1 of **B** against the 2·54 on the **A**, and then the number of inches on the **B** scale will be opposite the number of centimetres on the **A** scale. Thus we find—

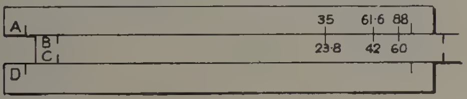


Fig. 8

2·9 in. = 7·33 cm. 9·5 in. = 24·1 cm.
15 cm. = 5·91 in.

The way in which these appear is indicated in Fig. 7.

Example 2. To convert m.p.h. to ft./sec. We know that 60 m.p.h. = 88 ft./sec., so we set 60 on the **B** scale against 88 on the **A** scale. With this setting we read

42 m.p.h. = 61·6 ft./sec., 35 ft./sec. = 23·8 m.p.h.

as indicated in Fig. 8.

To evaluate longer fractions, much time can be saved by dealing with the numbers in such an order that the fewest settings are used. The principle is to alternate, as far as possible, multiplications and divisions, and not to do all the multiplications first and then the divisions afterwards.

Example 3.

$$\frac{61{\cdot}5 \times 4{\cdot}28 \times 0{\cdot}017}{35{,}100 \times 44}$$

In such an example, we disregard the decimal points, while working, and put them in at the end. A very rough estimate will give us the order of magnitude of the answer. In the above, we get for such a rough estimate

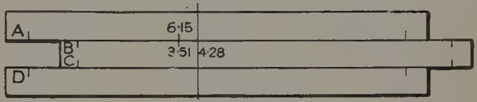


Fig. 9

$$\frac{60 \times 4 \times {\cdot}02}{30{,}000 \times 40} = 0{\cdot}000004$$

To use the rule, first divide 61·5 by 35,100, i.e. put 3·51 on the **B** scale opposite 6·15 on the **A** scale. Now to multiply by 4·28, slide the cursor along till the line coincides with 4·28 on the **B** scale. This is indicated in Fig. 9. (The reading on the **A** scale is, of course, $\frac{6{\cdot}15 \times 4{\cdot}28}{3{\cdot}51}$, but the value is not needed—the cursor "remembers" this for us.) Next, to divide by 44, bring 4·4 on the **B** scale under the line on the cursor, without moving the cursor, of course. The final answer is now read off on the **A** scale above 1·7—indicated in Fig. 10—it is 2·90, and the rough estimate above shows that this must be taken as 0·00000290, the value sought.

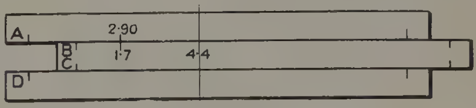


Fig. 10

In the above, it is seen that with one more factor in the numerator than in the denominator, there are just as many slide settings as there are factors in the denominator. Until practice has made one expert, it is worth while to put 1's in until there is one more factor in the numerator than in the denominator. For instance, think of

$$\frac{46{\cdot}5 \times 2{\cdot}17 \times 0{\cdot}085}{218} \text{ as } \frac{46{\cdot}5 \times 2{\cdot}17 \times 0{\cdot}085}{218 \times 1}$$

or of

$$\frac{512 \times 0{\cdot}97}{3{\cdot}71 \times 55{\cdot}2 \times 0{\cdot}46} \text{ as } \frac{512 \times 0{\cdot}97 \times 1 \times 1}{3{\cdot}71 \times 55{\cdot}2 \times 0{\cdot}46}$$

and apply alternately factors from the numerator and the denominator as in example 3.

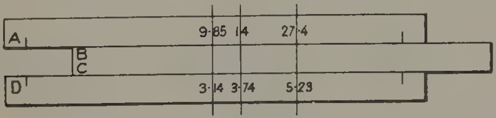


Fig. 11

Squares and Square Roots. The fact that the unit of the **A** and **B** scales is half that of the **C** and **D** scales has very useful consequences. It means that any number on the **A** scale is the square of the number directly opposite on the **D** scale (and similarly with the **B** and **C** scales), and conversely that the numbers on the **D** scale are the square roots of the numbers opposite them on the **A** scale. To read, we may either use the cursor, or the beginnings or ends of the **B** and **C** scales. A slide-rule is thus a portable table of squares and square roots.

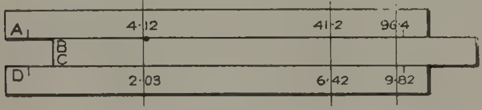


Fig. 12

Example 4. $5{\cdot}23^2$. Set the cursor line over 5·23 on the **D** scale, and read the answer, 27·4, on the **A** scale. (See Fig. 11.)

Example 5. 314^2. Set the line of the cursor on 3·14 on the **D** scale, and read 9·85 on the **A** scale. Since $300^2 = 90{,}000$, the value of 314^2 (to slide-rule accuracy) is 98,500. (See Fig. 11.)

Example 6. $\sqrt{14}$. Set the cursor on 14 of the **A** scale, and read the result, 3·74 on the **D** scale. (See Fig. 11.)

For numbers which do not appear on the **A** scale, such as 41,200, or 0·00964, care is needed to use the correct portion of the **A** scale —that between 1 and 10 or that between 10 and 100. By using the rule of marking off in pairs from the decimal part, as is done in extracting square roots by the arithmetical process, the correct setting can be obtained.

Example 7. $\sqrt{41{,}200}$. Mark off in pairs thus, 4, 12, 00, and now set the cursor on 4·12 (i.e. consider the first mark of division from the left as the temporary decimal point), on the **A** scale, and read on the **D** scale 2·03; putting the decimal point in its correct position, the answer is 203. (See Fig. 12.)

Example 8. $\sqrt{4120}$. Marking off in pairs, we have 41, 20. Set the cursor on 41·2 of the **A** scale, and read 6·42 on the **D** scale. Placing the point, the answer is 64·2. (See Fig. 12.)

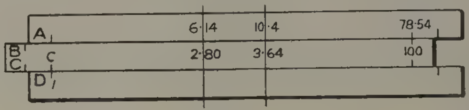


Fig. 13

Example 9. $\sqrt{0{\cdot}00964}$. Marking off in pairs, we have 0·00, 96, 4. Set the cursor on 96·4 of the **A** scale, and read 9·82 on the **D** scale. With the point placed, the result is 0·0982. (See Fig. 12.)

The combined use of the two pairs of scales will enable us to find very quickly the values of fractions containing squares and square roots. A very common case is provided by calculations on areas of circles.

Example 10. Find the area of a circle of diameter 3·64 in. Here $A = \frac{\pi}{4} \times 3{\cdot}64^2$. Set the 100 of the **B** scale against 78·54 $\left(\frac{\pi}{4} = 0{\cdot}7854\right)$ of the **A** scale, slide the cursor until the line lies above 3·64 on the **C** scale, and read the answer 10·4 sq. in. on the **A** scale. (See Fig. 13.) In doing this, it will be noticed that there is an extra graduation marked *C* (to denote *circle*) on the **C** scale, which lies now opposite the 1 of the

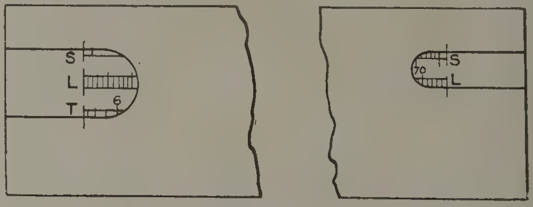


Fig. 14

D scale, as indicated in Fig. 13. The use of this graduation saves time in setting. With this setting, the area of any circle can be read off by sliding the cursor to the diameter on the **C** scale and reading on the **A** scale. Conversely, the diameter of any circle whose area is given can be read on the **C** scale by sliding the cursor to the area on the **A** scale. The correct portion of the **A** scale must be used, but the rule is exactly the same as that for extracting square roots. Thus, if the area is 0·0614 sq. in., marking off in pairs, gives 0·06, 14. Setting the cursor on 6·14 of the **A** scale, we read 2·80 on the **C** scale as indicated in Fig. 13, and the diameter is 0·280 in.

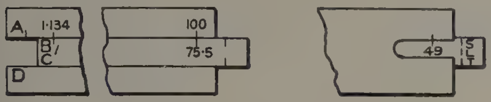


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FIG. 15

It will be seen that when a square occurs in the calculation, the **C** scale is used for it, but the **A** and **B** for other numbers (not squared). Conversely, if a square root occurs, the **B** scale is used, and the **C** and **D** for numbers outside the square root.

EXAMPLE 11. $h = \dfrac{0{\cdot}014 \times 250 \times 3{\cdot}5^2}{32{\cdot}2 \times 1{\cdot}25}$. Set 3·22 on **B** against 1·4 on **A**; slide cursor to 25 on **B**; now put 12·5 on **B** under cursor line; slide cursor to 3·5 on **C**, and read the result, 10·68 on **A**. A rough estimate shows the answer to be about 1, so $h = 1{\cdot}068$.

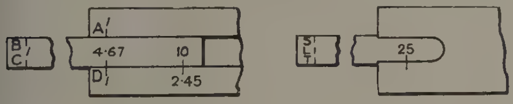

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FIG. 16

EXAMPLE 12. $T = 2\pi\sqrt{\dfrac{0{\cdot}45}{32{\cdot}2}}$. Cursor line on 6·28 ($2\pi$) on **D**; 32·2 on **B** under cursor line; slide cursor to 45 (mark off in two's from point) on **B**; read 7·41 on **D**. Placing the point by a rough estimate, we have $T = 0{\cdot}741$.

Back of Slide. In the better slide-rules there will be found three more scales on the back of the slide. They are labelled in the order *S*, *L*, and *T*, and enable us to find sines, logarithms, and tangents respectively. Index marks are engraved on the stock, as shown in Fig. 14. These allow readings to be taken without drawing out the slide completely and putting it in the other way. To find the *sine* of an angle, say, sin 49°, draw out the slide until 49 on the *S* scale is opposite the index mark. Turn over, and opposite 100 on the **A** scale we read 75·5 on the **B** scale. Putting the decimal point in the right place, this means that sin 49° = 0·755. (See Fig. 15.) (At the same time, above 1 on the **B** scale, we read 1·324 on the **A** scale; cosec 49° = 1·324.) To find the *cosine* of an angle, we look up in this way the sine of its complement, i.e. to find cos 38°, look up sin (90° − 38°) or sin 52°. To look up the *tangent* of an angle less than 45°, say, tan 25°, draw out the scale until the 25 on the *T* scale is opposite the index mark on the back of the stock. (See Fig. 16.) Turn over, and opposite the 1 on the

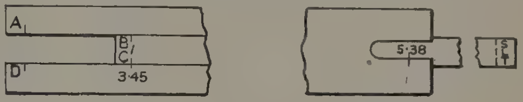

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FIG. 17

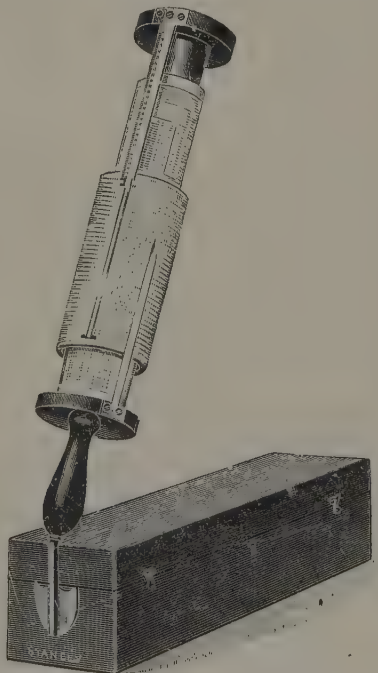

FIG. 18. FULLER SLIDE-RULE

D scale, read 4·67 on the **C** scale. This means that tan 25° = 0·467. At the same time, under 10 on the **C** scale we read 2·145 on the **D** scale. Cot 25° = 2·145. To find the tangent of an angle greater than 45°, find by this means the co-tangent of its complement. To find the *logarithm* of a number—its decimal part or mantissa, that is—set the 1 of the **C** scale opposite the number on the **D** scale, turn over and read the log. on the *L* scale, opposite the index mark. Thus, log. 3·45 = 0·538, as indicated in Fig. 17. By taking out the scale, and putting it in the other way, so that the *S* scale is next the **A** scale, the values of sines, tangents, and logarithms can be read directly (the last with the aid of the cursor).

Space does not permit more than a brief mention of other features. Some rules have a log-log. scale, which enables fractional powers of numbers to be obtained. Others have a scale with one-third the unit of the **C** scale, enabling cubes and cube roots to be read off. There are also special markings for special technical purposes, and cursors with two or three lines. A circular slide-rule, about the size of a watch, is also made. For greater accuracy, too, a long scale is wound helically round a cylinder, in the Fuller pattern shown in Fig. 18, the block for which was kindly lent by W. F. Stanley & Co. With a clear understanding of the principle, however, these other forms are as easy to master as the commoner straight rule.

(CONCLUSION)

TRAINING OF MECHANICAL ENGINEERS

By A. P. M. FLEMING, C.B.E., M.Sc., M.I.E.E.
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LESSON IV

WORKS TRAINING OF "PROFESSIONAL" ENGINEERS

TECHNICAL training is of paramount importance to the engineer, and it should be the first concern of the young man wishing to become an engineer to obtain technical training during a period of whole time day instruction.

At the conclusion of the university course, a period of practical training of not less than two years is required. This, preferably, should be spent in a manufacturing works which is large enough to embrace all the principal branches of engineering manufacture. During the period of practical training the student will spend short periods in various manufacturing and testing departments, and also obtain experience in the drawing office, design, commercial, works administration and research divisions. During this time the student should develop a bent for the work conducted in one or more of these divisions, and may eventually secure a junior position in one of them. While he is in the works departments he will be expected to work the same hours and under the same conditions as the workmen, and during any time he spends in the offices he will be treated as a junior member of the staff. The following gives a typical syllabus of training on a two year mechanical engineering "College Apprenticeship" course, the first year being spent in general workshop practice, and the second in more specialized training.

Training During the First Year—

(i.)	Pattern-shop and foundry .	3 months
(ii.)	Machine-shop . . .	4 ,,
(iii.)	Fitting and erecting . .	5 ,,

(i.) *Pattern-shop and Foundry.* During the time spent in the pattern-shop, instruction should be provided in the following subjects—

1. Factors in pattern-making determined by foundry practice. Function and character of loose pieces and their drawbacks.
2. Necessity for taper and provision for contraction.
3. Warping and shrinkage.
4. Application of cores and construction of core boxes.
5. Use of wood and metal patterns.
6. Economic application of pattern-making machinery.
7. Principles of construction of typical patterns, for example, turbine cylinder, condenser shell, generator yoke, generator bedplate, etc.

In the foundry, instruction should be given in the following subjects—

1. Green sand and dry sand moulding. The relative use of "bedding in" and "turning over" methods.
2. Venting. Risers and gates.
3. Loam moulding. Relative use of loam and dry sand moulding.
4. Plate and machine moulding. Considerations which determine their adoption.
5. Core-making. Dry sand and oil sand cores.
6. Cupola practice. Types of irons. Mixing. Blast. Fluxes.
7. Elementary considerations of the founding of non-ferrous metals.

(ii.) *Machine-shop.* In the machine-shop, nstruction should be provided in the following subjects—

1. Types of machine tools; their design and application, being a consideration of the more mportant machine tools and their relative uses.
2. Machine setting. The placing and securing of the work in the machine.
3. Cutting tools. Their form, cutting angles, setting in machine.
4. Tool layout. Tools required. Positioning and sequence.
5. Feeds and speeds.
6. Machine-shop requisites for repetition work. Jigs, form tools, attachments and ccessories.
7. Machine-shop costs. Special features. Machine hour. Distribution of overhead harges.

(iii.) *Fitting and Erecting.* In a factory producing steam turbines, pumps, condensers, etc., xperience in mechanical fitting and erecting would relate to these machines and would deal with the following subjects—

1. Use of tools. Chisels, files, and scrapers.
2. Principles of minor assembly, with special eference to such details as nozzle boxes and alve gear. Keys and keyways.
3. Measurements and measuring instruments. Rules, gauges, micrometer, vernier.
4. Levelling of bedplates and lining-up of ylinders and bearings.
5. Bearings, bedding-in. Allowances for lignment. Common troubles. Lubrication. alancing.
6. General shop layout. Use of cranes. linging.
7. Production and costing systems as applied to fitting and erecting shops. Overhead expenses. Methods of wage payment.
8. Distribution of responsibilities between foremen, inspectors, and workmen.

Second Year Training. In a manufacturing works the second year course of practical training for university trained engineers is intended to equip students to undertake employment along one of the following lines—

(i.) *Works Organization.* This comprises the study and application of economic methods of production. An important branch of this work relates to the erection of apparatus either for land or marine purposes.

(ii.) *Commercial Work.* This relates to the organization of home and foreign sales of manufactured apparatus. The training for commercial work comprises experience in testing, erection, and tender departments, and, in addition to work in these branches, some knowledge must be obtained of selling methods in relation to industrial plant, contracts, and correspondence, letter writing, conditions of sale, considerations of the law of contracts, credit and its approval, and the relative characteristics of home and export sales.

(iii.) *Engineering Design.* Experience in this direction relates to the design and calculation of apparatus, and the preparation of technical information. In general, the training for engineering design is concentrated in the test departments, drawing offices and engineering design offices. As a rule, the drawing office experience reveals an engineer's suitability for this particular type of work.

(iv.) *Research.* This type of work demands special characteristics, and is only available to those engineers possessing a very sound knowledge of fundamental scientific principles. Physics and mathematics are essential tools to the research engineer.

(v.) *Accounting.* This work includes general accounting and cost accounting, and particularly in the latter, a good general knowledge of manufacturing processes and methods is a valuable asset. Where college apprentices desire to undertake this type of work, it is advisable that, in addition to their engineering qualifications, they should obtain membership of a recognized professional institution relating to accounting work, for example, the Institute of Chartered Accountants, or the Institute of Works and Cost Accountants.

It is during the second year in a large works that college apprentices get the best opportunities for judging in what particular direction they are most likely to succeed, and when a man is sure of his bent he should direct his attention more specifically to preparing for a junior position in this sphere. He should not, however, fail to appreciate the inter-relationship that exists between his and all other engineering branches. It is increasingly the practice for large concerns to draw their junior staff from the ranks of their apprentices, and usually before the end of his period of training, a man will have a fairly clear understanding as to the possibilities of securing a suitable opening at the conclusion of his apprenticeship course. This is one of the reasons why it is best to finish training in a works rather than in a university, as during his period in works an apprentice is more likely to come in contact with suitable opportunities for future employment.

One of the best stepping-off grounds for employment in a variety of engineering directions is from erection work. Most large firms give their erection engineers a special erection course, usually of 12 months' duration. Qualifications in three main directions are required for erection engineers; practical training of a composite character; technical training of a first-class order; and organizing ability.

This training provides adequate facilities for promotion for the best type of man, and offers opportunities not only in the manufacturing organization concerned, but also as engineers to customers, and in numerous other directions including employment overseas.

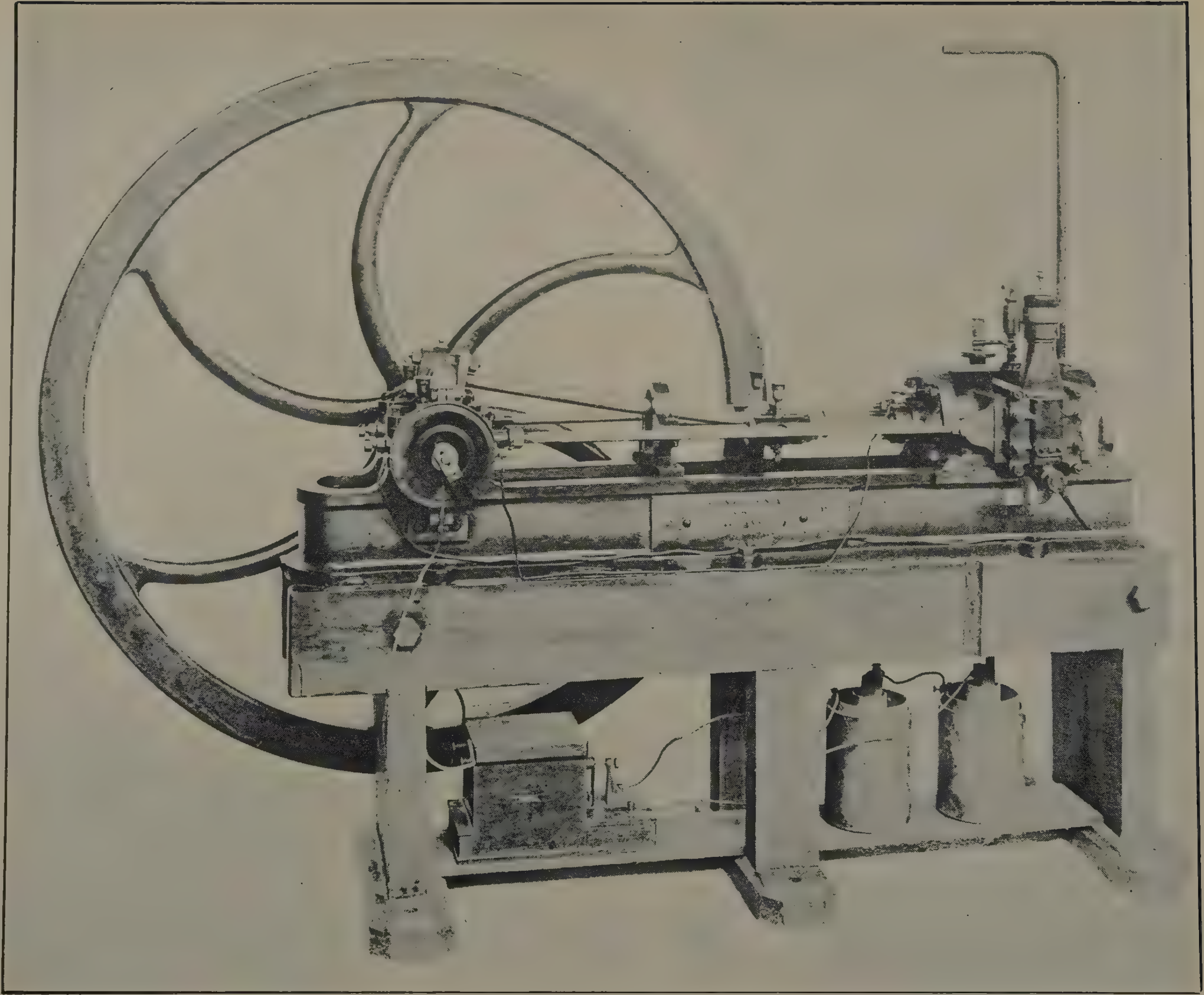
Wages During Practical Training. During the training of apprentices for artisan employment, wages dependent on the rates prevailing in the locality are paid. These are usually about 12s. per week at 14 years of age rising to about 32s. per week at 20. In the case of school and college apprentices the old method of premium paying has, to a large extent, been replaced by the payment of maintenance grants by the employers. These vary from 25 to 50s. per week.

Scholarship Facilities for Engineering Training. There is nowadays no reason why any boy possessing the necessary ability, no matter what his social and financial circumstances, should not obtain the education and training necessary to become a fully qualified technical engineer. Scholarship facilities now exist whereby a boy may make his own way from an early age. The number of free places in Central, Junior Technical, and Secondary Schools is increasing, and in very many cases maintenance grants are obtainable in connection with these. From the Secondary Schools facilities exist by which a student can proceed to the university and other higher educational institutions. Details and conditions of such scholarships can be obtained by application to the local educational authorites and the schools and colleges concerned. The scholarships of outstanding merit in engineering are the "Whitworth" and the "Royal," details of which can be obtained from the "Syllabus of the Science Scholarship Examination" and "Regulations for Whitworth Scholarships" respectively, issued by H.M. Stationery Office and obtainable through any bookseller. The British Electrical & Allied Manufacturers Association provides both electrical and mechanical engineering scholarships for youths who have "commenced a systematic course of training in engineering, but whose further training is stopped or impeded owing to the lack of the necessary funds." Details of these scholarships are obtainable from the B.E.A.M.A., 36 Kingsway, London, W.C.2.

Although fees differ in technical colleges and universities dependent on the extent to which they have been endowed, the cost of technical training is from £30 to £45 per yearly session with an additional £10 for instruments and books, etc. Maintenance costs in residential colleges are, of course, not included in these fees.

Personal Characteristics Required by Engineers. The important part which engineering plays in the industrial life of the nation indicates the importance that is attached to the necessity for absorbing only the very best human material into the industry. In selecting applicants for posts in the technical branches of the work, the industry looks at the inherent characteristics of the applicant first, and secondly, at his academic attainments. The qualities of character that help to make a successful engineer are independence of thought, a sense of responsibility and self-reliance, a capacity to get things done, an intense appreciation of practical needs, ability to deduce cause from effect, and to face a new situation without having to consult precedent, a team spirit and an outstanding personality. Academically, a great deal of weight is attached to mathematical ability, a natural aptitude for physical science and the ability to express oneself in good English.

(CONCLUSION)



By permission of *the Director of the Science Museum*

FIG. 23. LENOIR GAS ENGINE, 1860
(See page 565)

FILING TECHNICAL JOURNAL INFORMATION

By J. Swinburne, F.R.S.

OST engineers take in some half-dozen technical pers, and the journals of several societies. It impossible to remember all that one reads ; d numerical information must either be lost, filed in some way so that it is available when anted.

Only a small portion of any journal concerns y one particular enneer, and as journals d societies are multiying like germs, the fficulty gets worse. o keep periodicals in undles or packets is use ; and to bind em is expensive, and volves too much ookcase room. One fficulty is that jourals are all different zes. Many years ago British association ommittee tackled the uestion, and peraded several of the ocieties to bring their urnals within certain mits of letterpress d margin, so that arious papers or exracts could be bound ogether. Many atempts have been ade to get catalogues niform in size. Except in the case of pubshers, these attempts ave failed. Catagues are a serious ouble. They are so untidy in shelves, and eir unevenness wastes so much room that any an expensive catalogue of oblong or other lly shape goes straight into the waste-paper asket.

J. Swinburne, Esq., F.R.S.

One method of keeping track of information to cut the pages out, and file an article under e author's name. It is curiously easy to emember an author's name ; while there is a ifficulty in filing under subject, as a paper often deals with several subjects, or is in a borderland, so that it might be filed under one of several heads. This method is useful only for an individual ; it cannot work in an office where several are concerned, as they may not have seen the article to begin with, and therefore cannot think of the author's name.

Another system is better. Thus an index of subjects is made with numbers or decimals ; so that new subjects, or subdivisions can be added. This index is kept on one's desk. An article that is worth keeping is then marked with its number in pencil. An office flapper looks through the journal and cuts out the marked articles and files them under their numbers. She may card-index the authors' names, too ; but that is extra work. This system works quite well. The numbers may be on the lines of those used by the Royal Society. There is, I believe, an international system of numbers, and if so it is best to use it ; though a private system works all right. The drawback to this system is that in ten to twenty years the collection gets bulky, and much of the material is obsolete. Not only are shelf room and filing boxes expensive, but post-war flappers are much dearer, and they call themselves secretaries, and do not flap so well.

Cutting out articles and pasting them into books is quite useless. It involves having two copies of each journal, or else splitting the paper. Splitting paper is much easier than might be thought. The article is pasted on both sides, and a sheet of thin tough manilla paper or vegetable parchment stuck on each side. When dry the manilla sheets are pulled apart, and the article splits, half going with each manilla sheet. The split sheets are then soaked off and mounted.

There is another method of keeping track of technical information, and that is using published indexes or indices. There is the *Engineering Index*, and there is or was a very complete index in three languages which came out at short intervals. This involves access to a public library. Even in Westminster, or near the Patent Office library this is inconvenient, and many engineers cannot easily consult back numbers. One or two of the published indexes have a very good system, by which subscribers can, for a small sum, get any of the articles indexed.

Science Abstracts is most valuable to physicists, electrical and power engineers, and a long file takes little room and is a mine of information. It does not take long to run through *Science Abstracts* every month, and the *Engineering Abstracts* of the Institution of Civil Engineers.

There is something very attractive about having a complete system of filing, so that all information is available just when you want it. But this may be illusory. The Royal Society catalogue of papers seemed indispensable to every scientific worker, but it came to an end because there was not enough demand for it. Experience of many years of filing has shown me that it is not always worth the trouble. The occasions on which the file is used are extraordinarily few, and the group of papers on any subject is always partly obsolete, and rather incomplete; so that it is much better to get the latest textbook on the subject. One can go farther and rely entirely on *Science Abstracts* and *Engineering Abstracts*, and stop all technical papers, and all societies' journals that can be stopped. This may be very bad advice to a young man, and it ought to mean increasing ignorance and stagnation; but it does mean much happiness. After all, the abstracts do keep a reader up to date to some extent. What is wanted is a glorified system of abstracts in which all the technical and scientific societies collaborate. A subscriber would take in those of the subjects that interest him. The abstracts would then be practically complete; and there would be no duplication. Various attempts have been made to get the societies to combine as to abstracts, as to libraries, and as to buildings; but these attempts always failed. It may be said that if every one read abstracts only, there would be no journal readers, and no journals. But most engineers are specialists to such an extent that one or two journals cover the subjects in which each is interested; so the demand for engineering papers is not likely to go down. But any attempt at keeping abreast of the broad subject of engineering by studying from six to a dozen journals a week is hopeless.

Another source of information that is much neglected is patent literature. You can get abridgements of patents of any class for 6d. a year for past years; but a further subscription gets the sheets of abstracts as they are printed, so that you can keep abreast with invention very easily.

For any engineer who is something of a specialist, and who has a clear idea of the information he is likely to want in the future, the simple plan of pencilling a number on anything likely to be wanted and having a series of cases works very well. On looking up any subject the obsolete matter can be removed and thrown away, so that after some time the collection does not grow very rapidly.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XI

he Finished Detail Drawing. We have so far onfined our attention to the making of lines epresenting form, together with dimensions iving the size of the various sections of the achine part. To complete the drawing for he workshops it is necessary to provide further nformation regarding material, quantities, and ractice to be adopted in making the part. As e have previously remarked, the finished draw- g should be capable of providing the craftsman rith all the information required in the manu- acture of the part. Consequently, notes are equired on the finished working drawing and, order to obtain some consistency in style, it ay be advisable to give some attention to the ractice of lettering machine drawings.

A B C D E F G H I J K L M

N O P Q R S T U V W X Y Z

1 2 3 4 5 6 7 8 9 0.

Fig. 1

Lettering. In examining the modern style machine drawing, the obvious note is legi- ility independent of beauty. The purpose of e lettering is to give clearness to the instruc- ons rather than embellishment to the drawing. urther, as most drawing offices issue photo rints, taken from ink tracings of the drawings, the workshops, it is only necessary to make e lettering on the drawing legible to the tracer.

We do not, from these considerations, urge slovenly style, but rather that the best style adopt is one which gives legibility with the ast effort and time expenditure.

The utilitarian aspect of this question has ecently been investigated by an American Electric Company" employing hundreds of raughtsmen and, as a result of changing the yle of lettering from the vertical to the slant, ey claim to have saved about 25 per cent of e time taken on this part of the drawing.

Adopting the above style as shown, Fig. 1, e may consider the method of making the tters. Until practice provides the necessary dgment of space we suggest making the letters between lines spaced $\frac{1}{8}$ in. apart for small notes and $\frac{1}{4}$ in. apart for titles. When practice has given some facility in lettering, the guidance lines may be replaced by lettering immediately above the set square, as shown in Fig. 2.

Purpose of Assembly. We have, in a previous lesson, remarked upon the value of showing the details of a machine part assembled in order to give the craftsman an idea of the finished part. Another important purpose to be served in detailing the parts assembled is to check the dimensions of the adjoining parts. Further, as the construction of the machine part proceeds, many of the dimensions are automatically fixed to correspond to the adjacent part, rather than determined by either calculation or rule of proportion.

List of Parts. Fig. 3 shows a detail drawing of a relief valve for a hydraulic pump of moderate pressure. The list of parts given on the drawing gives a name or description of the various parts together with the material, number required, and any remarks peculiar to the part. It will be observed that a distinguishing letter is allocated to each part and placed on or against the part in the drawing. Corresponding notation is provided in the list of parts, so that the reader may readily find the particulars of any specified part.

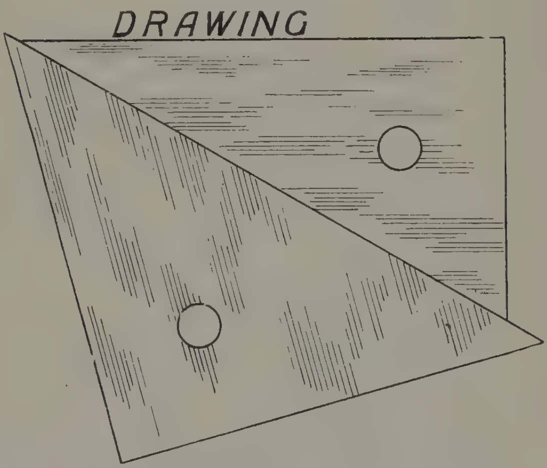


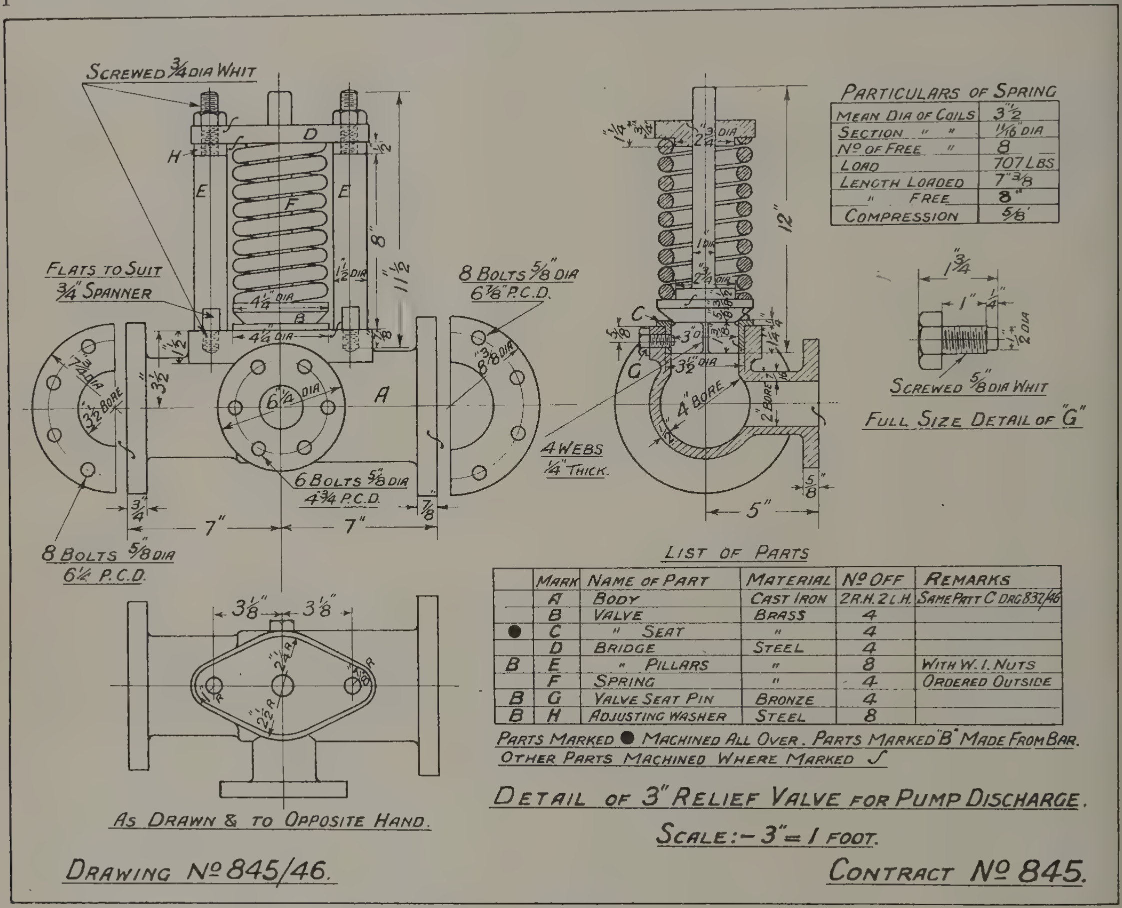
Fig. 2

The column in the list set apart for remarks is usually utilized for instructions to the pattern-maker, regarding the handing of the casting or the existence of a pattern which may be used. Notes applicable to all parts are placed at the foot of the list and are usually of a common character, such as a reference to the machining required on the various parts.

Handing. It will be seen (Fig. 3) that the number of relief valves required are two, as drawn, and two of opposite hand. Although this note is of a general nature, it evidently applies only to those parts which are not symmetrical. In this instance all parts are symmetrical with the exception of the body *A*, the note therefore refers to this part only. The alternative outline of the body *A* is shown in Fig. 4, and this form of body is intended in the reference *to opposite hand.*

Fig. 5 shows a further example explaining the term *handing*. It will be evident from Figs. 6 and 7 that a difference in the relative position of the sections of a part does not always entail a difference in the form of the part, but simply a re-arrangement relative to the adjoining part.

Machining. Whilst practice usually decides the question of the amount of machining required on a machine part, the craftsman becoming as familiar as the draughtsman regarding this requirement, the responsibility of deciding the machining is left with the draughtsman. As a general rule, the degree or finish of the machining is left to the judgment of the shop foreman, and only in new or special cases is the nature or degree of the machining specified on the drawing. The method shown in Fig. 3 has the dual



PARTICULARS OF SPRING

Mean Dia of Coils	3"½
Section " "	11/16" dia
Nº of Free "	8
Load	707 Lbs
Length Loaded	7"⅜
" Free	8"
Compression	⅝'

LIST OF PARTS

	Mark	Name of Part	Material	Nº Off	Remarks
	A	Body	Cast Iron	2 R.H. 2 L.H.	Same Patt C Drg 832/46
	B	Valve	Brass	4	
●	C	" Seat	"	4	
	D	Bridge	Steel	4	
B	E	" Pillars	"	8	With W.I. Nuts
	F	Spring	"	4	Ordered Outside
B	G	Valve Seat Pin	Bronze	4	
B	H	Adjusting Washer	Steel	8	

Parts Marked ● Machined All Over. Parts Marked "B" Made From Bar.
Other Parts Machined Where Marked ∫

DETAIL OF 3" RELIEF VALVE FOR PUMP DISCHARGE.

SCALE:– 3" = 1 FOOT.

Fig. 3

advantage of being quickly added to the drawng and of not requiring any special addition to the workshop prints. The alternative method shown in Fig. 6, whilst taking more time to add to the drawing, is more distinct, and gives the reader a quicker idea of the machining requirements. A second alternative is given in Fig. 7 and, whilst requiring more time to produce, being added to the prints, provides the clearest ndication of the machining required. Further, this last method has the advantage of enabling the draughtsman to distinguish, in the case of castings, between the need for a definitely good machined face for good fitting and a bare clean up for less important fitting. When the former s required, the machining marks are added to both the pattern shop and machine shop drawngs, and when the latter only is required the machining marks are added to the machine shop copy only.

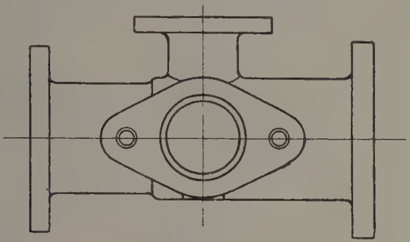
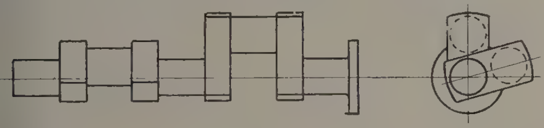
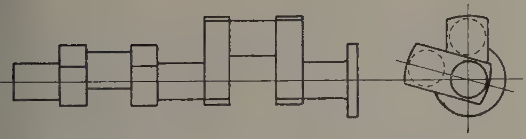


FIG. 4



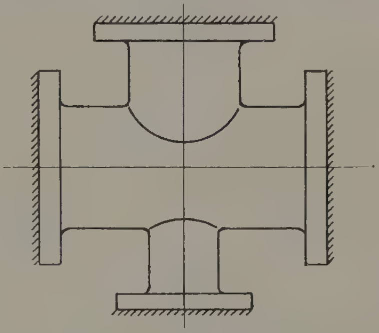
"RIGHT HAND" OR "AS DRAWN."



"LEFT HAND" OR "TO OPPOSITE HAND."

FIG. 5

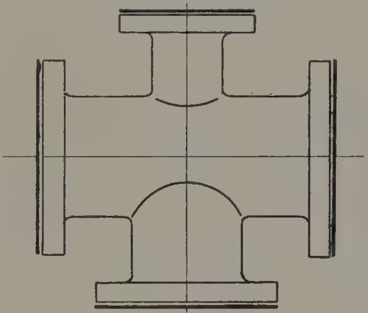
Title and Reference. The title which is usually placed in the right-hand bottom corner of the drawing should be brief and distinctive. For example, the title for the detail shown in Fig. 3 might have been "**Detail of 3 in. diameter pring-loaded relief valve for pump discharge,**" but the fact of it being a spring-loaded valve is self-evident, therefore this part of the proposed title may be considered redundant. The latter part of the above title giving the purpose of the detail is not evident from the drawing, and requires noting. So, for brevity and distinction, the title, "**Detail of 3 in. relief valve for pump discharge,**" is quite satisfactory.



MACHINED WHERE MARKED ////////

FIG. 6

To relate the detail to others for the same contract, a contract number is given to the drawing, and is usually placed below the title. Also the scale of the drawing is placed immediately below the title.



MACHINED WHERE MARKED IN RED
(Shown in thick line)

FIG. 7

For purpose of registration, a reference number is given to the drawing and is usually placed in the left-hand bottom corner. Where the drawing is for a particular contract the above number is preceded by the contract number.

EXERCISE

Practice the alphabet and numbers to a depth of $\frac{1}{8}$ and $\frac{1}{4}$ in., and later make a finished working drawing of a relief valve similar to Fig. 3, but to opposite hand.

APPLIED MECHANICS

GEO. W. BIRD, WH.EX., B.SC., A.M.I.MECH.E., A.M.I.E.E.

LESSON XII

BUILT-IN BEAMS

B.M. AND S.F. DIAGRAMS

WE have already considered the case of the *simply supported* beam, and drawn the B.M. and S.F. diagrams for both concentrated and distributed loading. The case of the *built-in* beam is obtained by applying fixing moments to the ends and, as a result, the whole beam becomes subject to a bending moment of an opposite kind to that produced by the direct loads carried by the beam. As generally arranged, the direct loads will tend to make the beam bend concave, as seen from above, whilst the end fixings will tend to make the beam concave as seen from below.

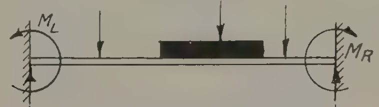


FIG. 58

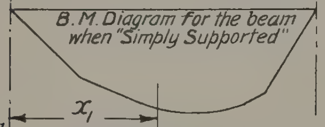


FIG. 59

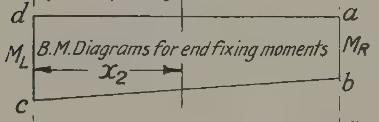


FIG. 60

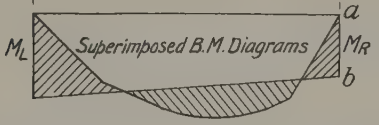


FIG. 61

Let Fig. 58 represent a beam carrying any system of loads as shown, and having its ends fixed horizontally by fixing moments M_L and M_R. The B.M. diagram for the *simply supported* state is represented in Fig. 59, whilst the B.M. diagram for the end fixing moments is the figure *abcd*, Fig. 60. The B.M. diagram giving the actual B.M. on any section of the beam is obtained by superimposing these two diagrams, Figs. 59 and 60, as shown in Fig. 61.

Let us consider the case of a beam which is *built-in* horizontally at its ends, and carries any type of loading; E and I being constant. We have to establish two relations between Figs. 59 and 60, viz.,

1. The area A_1 of Fig. 59 = the area A_2 of Fig. 60; and
2. The CG of A_1 lies immediately above the CG of A_2, or $x_1 = x_2$.

We found in Lesson IX, that if we integrate the algebraic expression for $\frac{M}{EI}$ we obtain the expression for the slope; and, further, if we perform this integration over the whole length of the beam, that is, from $x = o$ to $x = l$, we obtain the total change of slope over the whole length of the beam. Now, since the ends of the beam are fixed horizontally, the total change of slope over the whole length of the beam is zero, and hence we may write (assuming E and I are constant)—

$$\int_{x=o}^{x=l} (M_1 + M_2)\, dx = O$$

$$\text{or} \quad \int_{x=o}^{x=l} M_1\, dx = -\int_{x=o}^{x=l} M_2\, dx$$

where

M_1 = B.M. due to the loading on the beam.
M_2 = B.M. due to the end fixing moments.
= length of beam.
x = distance of the section considered from the origin.

But $\int_{x=o}^{x=l} M_1 dx$ = the area of the B.M. diagram, due to the loading, that is Fig. 59 = A_1.

Also $-\int_{x=o}^{x=l} M_2 dx$ = the area of the B.M. diagram, due to the fixing moments, that is, Fig. 60 = A_2.

Hence, we see that $A_1 = A_2$.

Again, we have shown in Lesson IX that

$$M = EI\frac{d^2y}{dx^2}; \text{ where } y = \text{deflection.}$$

Now, at any section whose distance is x from the origin (one of the fixed ends), the resultant B.M. $= M_1 + M_2$; hence we may write

$$\frac{d^2y}{dx^2} = \frac{1}{EI}(M_1 + M_2) \qquad . \quad (1)$$

Multiplying equation (1) by x, we get

$$x \cdot \frac{d^2y}{dx^2} = \frac{1}{EI}(M_1 + M_2)x \qquad . \quad (2)$$

Equation (2) can now be integrated by the aid of the rule for "Integration by Parts," and becomes

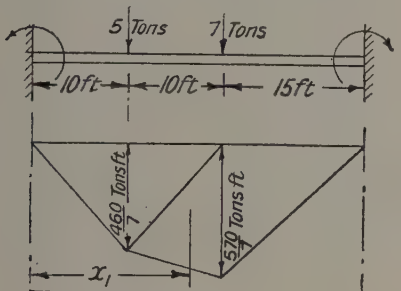


FIG. 62

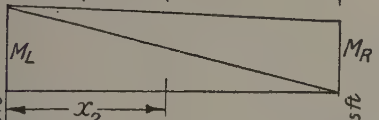


FIG. 63

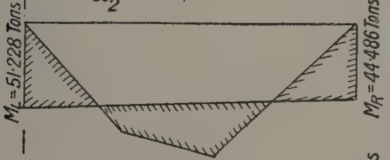


FIG. 64

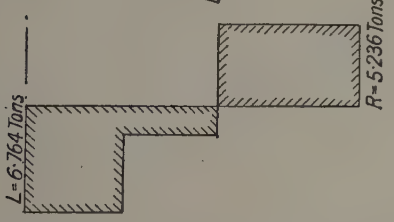


FIG. 65

$$\left[x \cdot \frac{dy}{dx} - \int \frac{dy}{dx} \cdot dx\right]_{x=0}^{x=l}$$

$$= \frac{1}{EI}\int_{x=0}^{x=l}(M_1 + M_2)x.dx.$$

or

$$\left[x \cdot \frac{dy}{dx} - y\right]_{x=0}^{x=l}$$

$$= \frac{1}{EI}\left[\int_{x=0}^{x=l} M_1\, x \,.\, dx + \int_{x=0}^{x=l} M_2 x \,.\, dx\right] \qquad . \quad . \quad . \quad . \quad (3)$$

Now $\frac{dy}{dx} = O$ when $x =$ either l or O.

Also $y = O$ when $x =$ either l or O; hence the whole expression

$$\left[x \cdot \frac{dy}{dx} - y\right]_{x=0}^{x=l} = O; \ \therefore \text{(3) may be written}$$

$$O = \frac{1}{EI}\left[\int_{x=0}^{x=l} M_1 x \,.\, dx + \int_{x=0}^{x=l} M_2 x \,.\, dx\right]$$

Again, $\int_{x=0}^{x=l} M_1\, x \,.\, dx = A_1 x_1$

where $x_1 =$ distance of the CG of A_1 from the origin, which may be taken at either end;

And $\int_{x=0}^{x=l} M_2\, x \,.\, dx = A_2 x_2$

But we have shown that $A_1 = A_2$, $\therefore$ $x_1 = x_2$, that is, the C.G. of A_1 lies immediately over the C.G. of A_2.

EXAMPLE. A girder of 35 ft. span has its ends fixed horizontally and carries loads of 5 tons and 7 tons at 10 ft. and 20 ft. respectively from one end. Calculate the end fixing moments and draw the complete B.M. and S.F. diagrams.

We first treat the beam as being *simply supported* at its ends, and draw the corresponding B.M. diagram, Fig. 62.

Area of Fig. 62 $= A_1 = 1675$ in ft. ton units $= A_2 =$ area of Fig. 63.

Taking moments of areas about the left-hand support, we get—

$$A_1 x_1 = \left(\frac{2300}{7} \times \frac{20}{3}\right) + \left(\frac{2300}{7} \times \frac{40}{3}\right)$$

$$+ \left(\frac{2850}{7} \times \frac{50}{3}\right) + \left(\frac{4275 \times 25}{7}\right)$$

$$\therefore \ x_1 = \frac{601{,}125}{1675 \times 21} = 17{\cdot}1 \text{ ft. nearly} = x_2.$$

We can obtain an equation involving M_L and M_R from Fig. 63.

$$A_2 = \left(\frac{M_L + M_R}{2}\right) \times l$$

$$\text{or} \quad 1675 = \left(\frac{M_L + M_R}{2}\right) \times 35$$

M_R=fixing moment about right-hand support.
M_L= fixing moment about left-hand support.

$$\therefore \ M_R + M_L = 95{\cdot}714 \qquad . \qquad (1)$$

Also, another equation can be obtained by taking moments of areas of Fig. 63, about the left-hand support—

$$A_2x_2 = \left(\frac{35}{2} \times \frac{35}{3} \times M_L\right) + \left(M_R \times \frac{35}{2} \times \frac{2 \times 35}{3}\right)$$

substituting $A_2 = 1675$ and $x_2 = 17{\cdot}1$ and simplifying we get

$$2M_R + M_L = 140{\cdot}2 \quad . \quad . \quad (2)$$

Combining (1) and (2) and solving we get—

$$M_R = \underline{44{\cdot}486} \text{ tons ft.}$$

$$M_L = \underline{51{\cdot}228} \text{ tons ft.}$$

REACTIONS

Taking moments about the right-hand end, e get—

$$L \times 35 + 44{\cdot}486 = 105 + 125 + 51{\cdot}228$$

$$\therefore\ L = \underline{6{\cdot}764} \text{ tons.}$$

Again, taking moments about the left-hand end, we get

$$R \times 35 + 51{\cdot}228 = 44{\cdot}486 + 50 + 140$$

$$\therefore\ R = \underline{5{\cdot}236} \text{ tons.}$$

L = reactions at left-hand support.
R = reaction at right-hand support.

With the calculated value of M_R and M_L, the fixing couple diagram can be drawn superimposed on the B.M. diagram for the simply supported condition as shown in Fig. 64, thus obtaining the resultant B.M. diagram for the beam. The calculated reactions, L and R, enable us to draw Fig. 65, the S.F. diagram for the beam.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XII

THE STEAMSHIP MADE PRACTICABLE

HENRY BELL (1767-1830)

Towards the end of the seventeenth century, attention was once again turned, after the ill-fated experiments of Jonathan Hulls, to the possibility of steam navigation. At this period, the difficulties which Hulls had had to contend with in the way of imparting rotary motion to the paddle wheels of his boat had disappeared. Watt's invention of the rotary engine, combined with the smoothness and the degree of freedom from shocks and irregular movements which that engine gave, contributed very greatly to the comparative ease with which the steamboat inventors from this period onwards attained success with their endeavours.

The first partially successful steamship was the "Charlotte Dundas," constructed by Symington in 1801. This was used as a tug on the Forth and Clyde canal, but, after a short time, its use was abandoned.

Other inventors, however, had taken up the subject of the steamship, more especially (in Great Britain) one Henry Bell, a native of Linlithgowshire. Bell had received very little education in his youth, but he possessed a natural "bent" for things mechanical, and he managed to get himself taken on as an apprentice in an engineering works. A few years elapsed, after which time he set up for himself as an engineer. Failure quickly overtook him in this direction, however, with the result that he was forced to accept the occupation of a third rate joiner.

Nevertheless, Bell's grit, courage, and enthusiasm for his pet subject—the construction of steamboats—triumphed in the end. In 1812, finding himself in possession of an increased income and more favourable prospects, Bell had constructed a steamship 40 ft. long. This was propelled by paddles placed at each side of the vessel, and which were driven by a single-cylinder engine.

Bell's "Comet," for such was the name which the inventor gave to the vessel, was entirely successful. It was capable of steaming against

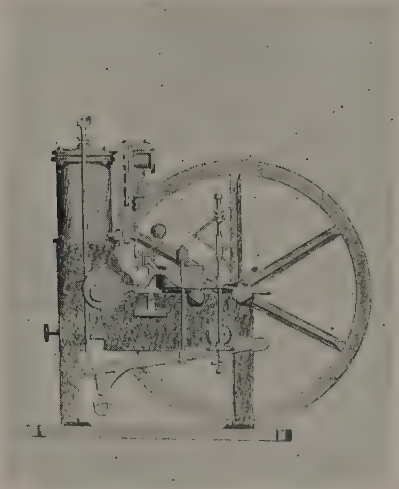
Henry Bell

The "Comet"
(From an Early Sketch)

a strong tidal current at the rate of seven miles per hour. The "Comet" was a success chiefly on account of the paddles being placed at the sides of the vessel instead of at the stern, as was he practice of the earlier steamboat construcors. Also, Bell's engine was a steady and mooth running one, and it was of a more robust onstruction than the earlier engines.

After August, 1812, the "Comet" made egular sailings between Glasgow and Greenock. t was, indeed, the first Clyde steamship, the rst of a long line of steam propelled vessels which have made the Clydeside world-famous as the home of steamship building.

The "Comet" in Full Steam


The Engine of the "Comet"

Bell died in 1830. He did not make a fortune out of his invention, but, nevertheless, the success of the "Comet" relieved him of his former pressing financial embarrassments, and he was able to spend his later years in circumstances of comparative ease.

Rival claims for the honour of inventing the steamship have been put forward, but there is certainly no doubt that Bell's "Comet" was the first practical steam sailing vessel which attained any measure of economic success.

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
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LESSON XII

QUADRATIC EQUATIONS

In earlier lessons we have seen how to solve simple equations in one or two unknowns, and also, in Lesson X, how to solve any equations for one unknown by means of graphs. Of course, a formula is preferable to a graph, if it can be had, and is not too complicated to use. In the case of quadratic equations, i.e. those which contain no higher power than the second (or square) of the unknown, it is possible to solve by calculation, and to obtain a formula for the roots. We proceed to show how this is done, and recommend the engineer to use the graphical method for anything worse (containing higher powers, that is).

As a first example, we take

1. $$x^2 + 4x - 12 = 0$$

If the x term were not there, we should take the 12 to the other side, and then take the square root. This idea is a clue to be followed up. If we can make a square *including* the $4x$, we shall be able to take square roots, and the rest will be easy. Look back at some of the examples on algebraic multiplication, and on factors, and see that we there learned that

$$(x + a)^2 = x^2 + 2ax + a^2$$

Now, starting with $x^2 + 4x$, what must be done to make a complete square? Comparing we see the first term is as we want it, the second, $4x$, will be $2ax$ if $2a = 4$, i.e. if $a = 2$. To complete the square, we must put in the third term, a^2, or in our case, 2^2. Thus

$$x^2 + 4x + 2^2 = (x + 2)^2$$

Similarly, starting with $x^2 + 10x$, we should have $2a = 10$, $a = 5$, i.e. 5^2 (or 25) to add, so that

$$x^2 + 10x + 5^2 = (x + 5)^2$$

If a is negative, the rule is just the same. It is

Add the *square* of *half* the coefficient of x.

Returning to our equation, and using this device,

$$x^2 + 4x - 12 = 0$$

Transposing, $x^2 + 4x = 12$

Completing square, $x^2 + 4x + 2^2 = 12 + 2^2$

(if we add 2^2 to one side, we must also add it to the other).

That is,

$$(x + 2)^2 = 12 + 4 = 16$$

We can *now* take the square root of both sides, and in doing so, we must remember that the square root may be either positive or negative, so that we shall obtain *two* values for x. In this case, the square root of 16 may be either $+ 4$ or $- 4$, so

either $x + 2 = 4$ or $x + 2 = -4$

so $x = \underline{2}$ or $x = -\underline{6}$

It may be verified that each of these values for x does satisfy the original equation, and it is instructive to draw a graph of the expression $x^2 + 4x - 12$, and see that it is zero for these two values of x only.

The other possibilities that may arise are: (*a*) the coefficient of x may not be exactly divisible by 2, this only introduces fractions; (*b*) the numerical square root may not work out exactly, then go to as many places as are needed; (*c*) the coefficient of x^2 may not be unity, then divide throughout by whatever it is, and we shall have a single x^2 to begin with. These points are illustrated in the following examples—

2. $$x^2 - 3x = 6$$

completing square $x^2 - 3x + 1{\cdot}5^2 = 6 + 1{\cdot}5^2$

or $(x - 1{\cdot}5)^2 = 6 + 2{\cdot}25 = 8{\cdot}25$

From tables, or by logs, the square root of 8·25 is found to be 2·872 . . . So we have, putting in the alternative signs,

either $x - 1{\cdot}5 = 2{\cdot}872$ or $x - 1{\cdot}5 = -2{\cdot}872$

so $x = \underline{4{\cdot}372}$ or $x = -\underline{1{\cdot}372}$

3. $$3x^2 + 2x - 7 = 0$$

transposing $3x^2 + 2x = 7$

dividing by 3, $x^2 + \frac{2}{3}x = \frac{7}{3}$

completing square $x^2 + \frac{2}{3}x + (\frac{1}{3})^2 = \frac{7}{3} + (\frac{1}{3})^2$

or, $(x + \frac{1}{3})^2 = \frac{7}{3} + \frac{1}{9} = \frac{22}{9}$

taking the square root,

either, $x + \frac{1}{3} = \frac{4{\cdot}690}{3}$ or $x + \frac{1}{3} = -\frac{4{\cdot}690}{3}$

so that, $x = \frac{-1 + 4{\cdot}690}{3}$ or $\frac{-1 - 4{\cdot}690}{3}$

$$= \underline{1{\cdot}230} \quad \text{or} \quad -\underline{1{\cdot}897}$$

EXERCISE No. 20

Solve the following quadratic equations, giving the roots to about four significant figures, when they cannot be exactly expressed—

1. $x^2 - 6x + 8 = 0.$
2. $x^2 + 4x + 3 = 0.$
3. $x^2 + 2x - 6 = 0.$
4. $x^2 - 3{\cdot}2x + 1{\cdot}6 = 0.$
5. $6x^2 + 5x = 1.$
6. $3x^2 - 8x = 3.$
7. $6x^2 - 6x + 1 = 0.$

Other Methods for Quadratics. A simpler method of solving a quadratic is sometimes available, that is, when it is possible to discover factors. Take

$$x^2 - 5x + 6 = 0$$

Factorizing, we have,

$$(x - 2)(x - 3) = 0$$

Now, if the product of two or more factors is zero, then *one* of them *must be zero*. Thus, the above is true when

either $x - 2 = 0$ or $x - 3 = 0$

i.e. when $x = 2$ or $x = 3$

The same argument can be applied to *all* equations, and the connection between roots and factors is important. We see that if $(x - a)$ is a factor, then $x = a$ is a root. We are also led to a connection between the roots and the coefficients. Writing the general quadratic equation in the form

$$ax^2 + bx + c = 0$$

and denoting the roots by α and β, we must have $(x - \alpha)$ and $(x - \beta)$ as factors,) i.e. the expression must be a multiple of $(x - \alpha)(x - \beta)$. To obtain the correct term in x^2, it is evident that the expression must be a times this. Thus, identically

$$ax^2 + bx + c = a(x - \alpha)(x + \beta)$$
$$= ax^2 - a(\alpha + \beta)x + a\alpha\beta$$

Comparing coefficients, we see that we must have

$$a(\alpha + \beta) = -b, \quad a\alpha\beta = c$$

so that

sum of roots $= \alpha + \beta = -\frac{b}{a}$

product of roots $= \alpha\beta = \frac{c}{a}$

These are important, and one of their uses is to enable us to find the second root if one is known. In particular, they enable us to find the smaller root more accurately in cases where one is much larger than the other, without evaluating a square root to a large number of places. For instance, if $x^2 - 10x + 1 = 0$, we find that

$$(x - 5)^2 = 24 \quad \text{or} \quad (x - 5)^2 = -24$$

Four figure tables give $\sqrt{24} = 4{\cdot}899$, and the roots,

$$x = 9{\cdot}899 \quad \text{or} \quad x = 0{\cdot}101,$$

the second of which is only correct to three figures. But here the product of the roots is 1, and one of them is 9·899. The other must then be 1 ÷ 9·899 = 0·10102 . . ., and since the larger root is correct to about 1 in 10,000, we can rely upon the fifth place as being correct.

Formula for Roots. We can apply the method of the first section to the general equation, $ax^2 + bx + c = 0$, and so obtain a formula that will give the roots of any quadratic.

Transposing, $ax^2 + bx = -c$

dividing by a, $x^2 + \frac{bx}{a} = -\frac{c}{a}$

completing square, $x^2 + \frac{b}{a}x + \left(\frac{b}{2a}\right)^2$

$$= -\frac{c}{a} + \frac{b^2}{4a^2}$$

that is, $\left(x + \frac{b}{2a}\right)^2 = \frac{b^2 - 4ac}{4a^2}$

or $x + \frac{b}{2a} = \frac{\pm\sqrt{b^2 - 4ac}}{2a}$

whence $\underline{x = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a}}$

the required formula, which should be known. Using it to solve $7x^2 - 11x - 17 = 0$, we have

$$x = \frac{-(-11) \pm \sqrt{121 - 4 \times 7 \times (-17)}}{2 \times 7}$$

$$= \frac{11 \pm \sqrt{597}}{14} = \frac{11 \pm 24{\cdot}43}{14}$$

$$= \frac{35{\cdot}43}{14} \text{ or } \frac{-13{\cdot}43}{14}$$

$$= \underline{2{\cdot}531} \text{ or } \underline{-0{\cdot}9596}$$

EXERCISE No. 20 (*contd.*)

Solve, using factors—
8. $x^2 - 8x + 15 = 0$.
9. $x_2 + x - 156 = 0$.
10. $2x^2 + x = 3$.

Solve by formula—
11. $8x^2 - x = 14$.
12. $x^2 + 22x = 665$.

"Imaginary" Roots. It sometimes happens that the quantity under the square root sign turns out to be negative, in which case it is not possible to find the square root, since every square is positive. If the equation arises out of a practical problem, this impossibility of finding a solution means that we have tried to do the impossible, such as trying to "put a quart into a pint pot." For the present the student had better leave it there, having tried to discover why the practical problem is insoluble. It is, however, instructive to draw a graph in such a case. If we try to solve $x^2 + 2x + 4 = 0$, our formula gives

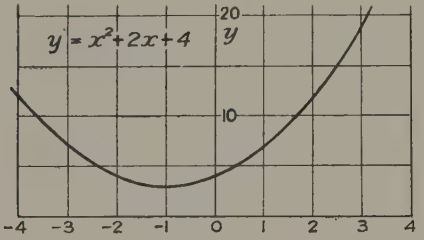


FIG. 26

$$x = \frac{-2 \pm \sqrt{-12}}{2}$$

which is impossible. Drawing a graph of the expression $x^2 + 2x + 4$ (see Fig. 26), we see that the expression never is zero, so that it is impossible to find roots.

BINOMIAL THEOREM

We have just been using the fact that

$$(x + a)^2 = x^2 + 2ax + a^2$$

There are also occasions upon which we need higher powers than the second of such a "binomial" expression (i.e. one with two terms). By continued multiplication it may be verified that

$$(x + a)^3 = x^3 + 3x^2a + 3xa^2 + a^3,$$
$$(x + a)^4 = x^4 + 4x^3a + 6x^2a^2 + 4xa^3 + a^4$$

and so on. It should be quite clear how the *indices* go in these "expansions," so that all we want is the law for the coefficients. This law was discovered by Sir Isaac Newton, and we give it without proof, as space will not permit more. As a matter of fact, it is usually needed as an approximation formula, the first two terms being enough. It is as follows—

$$(x + a)^n = x^n + nx^{n-1}a + \frac{n(n-1)}{1 \times 2} x^{n-2}a^2 + \frac{n(n-1)(n-2)}{1 \times 2 \times 3} x^{n-3}a^3 + \ldots$$

the fractions in the coefficients growing by one factor in numerator and in denominator each time, the numerator factors decreasing, and the denominator factors increasing, by 1 each time. When n is a whole number, the factors of the numerator will decrease to zero, and the series stops. If n is not a whole number, the series does not stop; it is still true, however, provided a is numerically less than x. As an instance,

$$(x + a)^7 = x^7 + 7x^6a + \frac{7 \times 6}{1 \times 2} x^5a^2 + \frac{7 \times 6 \times 5}{1 \times 2 \times 3} x^4a^3 + \frac{7 \times 6 \times 5 \times 4}{1 \times 2 \times 3 \times 4} x^3a^4 + \frac{7 \times 6 \times 5 \times 4 \times 3}{1 \times 2 \times 3 \times 4 \times 5} x^2a^5 + \frac{7 \times 6 \times 5 \times 4 \times 3 \times 2}{1 \times 2 \times 3 \times 4 \times 5 \times 6} xa^6 + \frac{7 \times 6 \times 5 \times 4 \times 3 \times 2 \times 1}{1 \times 2 \times 3 \times 4 \times 5 \times 6 \times 7} a^7$$

or, upon cancelling,

$$(x + a)^7 = x^7 + 7x^6a + 21x^5a^2 + 35x^4a^3 + 35x^3a^4 + 21x^2a^5 + 7xa^6 + a^7$$

It will be noticed that the coefficients of terms equidistant from the beginning and the end are equal, so that we need only calculate the coefficients half-way; this will be found to be true always.

A very useful form to remember is

$$(1+x)^n = 1 + nx + \frac{n(n-1)}{1{\cdot}2}x^2 + \frac{n(n-1)(n-2)}{1{\cdot}2{\cdot}3}x^3 + \dots$$

This is particularly useful when x is small, ince then the terms get rapidly less, and in nany cases the first two will give a sufficiently ccurate result. Thus

$$\begin{aligned} 1{\cdot}02^7 &= (1+0{\cdot}02)^7 \\ &= 1 + 7 \times 0{\cdot}02 + 21 \times 0{\cdot}0004 + \dots \\ &= 1 + 0{\cdot}14 + 0{\cdot}0084 + \dots \end{aligned}$$

Here the first two terms give $1{\cdot}14$, which is ess than 1 per cent in error. With a smaller

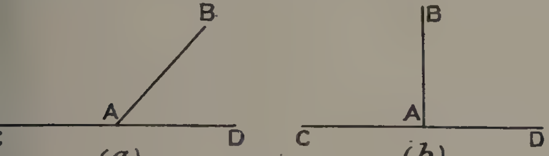


FIG. 27

ndex, or a smaller x, the accuracy is greater. Thus we may take

$$(1+x)^n = 1 + nx, \text{ when } x \text{ is small.}$$

As another instance of its use we will approxinate to $\sqrt{26}$. The answer is just over 5, and square root can be expressed by an index $\frac{1}{2}$. Then

$$\begin{aligned} 6^{\frac{1}{2}} &= (25+1)^{\frac{1}{2}} = \{25(1+0{\cdot}04)\}^{\frac{1}{2}} \\ &= 5\,(1+0{\cdot}04)^{\frac{1}{2}} \\ &= 5(1 + \tfrac{1}{2} \times 0{\cdot}04 \dots) = 5 \times 1{\cdot}02 \dots \\ &= 5{\cdot}10 \dots \end{aligned}$$

The value found by the use of five figure ables is $5{\cdot}0990$, so the error is about 1 in ,000 only.

If the sign in the bracket is –, this will mean hat *alternate* signs in the full expansion are –; nd for our approximation formula.

$$(1-x)^n = 1 - nx, \text{ when } x \text{ is small.}$$

EXERCISE No. 21

1. Expand $(x+a)^5$, $(1+x)^{10}$.
2. Use the binomial expansion to calculate $1{\cdot}02^5$ orrect to four places of decimals.
3. Obtain approximate values of $\sqrt{40}$, $\sqrt{97}$, $\sqrt[3]{30}$, y the use of the first two terms of the binomial xpansion.

GEOMETRY

The engineer has so many dealings with sizes and shapes of bodies, that no course of mathematics for him can be complete without some reference to geometry. We conclude this lesson by collecting together a few of the fundamental facts.

We take it for granted that the student is not altogether ignorant of lines and angles, and the use of drawing instruments, and of some of the more obvious geometrical facts.

Angle Properties. If one straight line, AB in Fig. 27*a*, stands on another straight line, CD, there are two angles between the lines, CAB and DAB. If we rotate AB about A, we shall reach a position (see Fig. 27*b*) when these two angles are *equal*. Each of them is then a *right angle*. It follows that the sum of *any* two angles made by one straight line standing on another, such as CAB and BAD in Fig. 27*a*, is *two* right angles, or 180°.

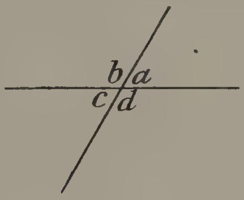


FIG. 28

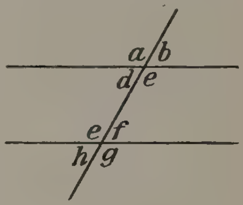


FIG. 29

If two straight lines cross one another, as in Fig. 28, we have four angles, and then the pairs of opposite angles are equal, i.e. $\angle a = \angle c$ and $\angle b = \angle d$. That this is so can be proved by use of the sum property above, for by it,

$$\angle a + \angle b = 180^\circ$$
$$\angle c + \angle b = 180^\circ$$

therefore, $\angle a = \angle c$

Parallel Lines. If a straight line crosses two *parallel* straight lines, as in Fig. 29, we get eight angles, which are equal in fours, i.e.

$$\angle a = \angle c = \angle e = \angle g$$

and $$\angle b = \angle d = \angle f = \angle h$$

Angles of a Triangle. It is a very remarkable fact that the angles of *all* triangles, no matter what shape they are, add up to two right angles or 180°. A fact so general as this cannot be *proved* by drawing a few triangles and measuring their angles, nor is it at all obvious. By use of the facts about angles and parallels given above, we may, however, show that it *must* be true. In the triangle ABC (Fig. 30), produce AC to D, and draw CE parallel to AB. Then, by the properties of parallel lines, the two angles marked x are equal, and so are the two marked y. Hence two important facts.

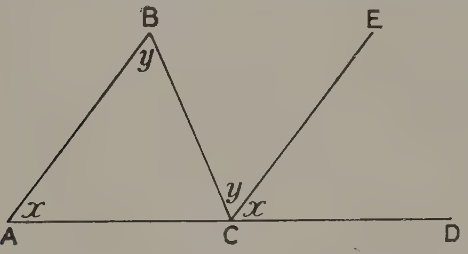


FIG. 30

1. The exterior angle of a triangle ($\angle BCD$ in Fig.) is equal to the sum of the two interior opposite angles (A and B).

Also, adding in the $\angle C$, we see that the sum of the three angles, A, B, and C, is equal to the sum of the three angles at C; these stand on a straight line, and therefore their sum is two right angles. Hence

2. The sum of the three angles of a triangle is two right angles or 180°.

ANSWERS TO EXERCISES NOS. 18 AND 19

(18)

1. $h = 0{\cdot}00117v^{1.825}$.
2. $H = 0{\cdot}0955v^{2.11}$.
3. $pv^{1.35} = 440$.

(19)

1. 7·75 ft./sec.
2. 7,300,000 gal., 3,630,000 gal.
3. 4,630 ft.-lb.

FITTING AND ERECTING

By JOSEPH G. HORNER, A.M.I.MECH.E.

LESSON VIII

THE ERECTION OF SHAFTING

THIS work differs in several respects from that which deals with those classes of erections that are self-contained, as say, engines, pumps, machines. The principal fact is that so much of it has to be done apart from lines and centres, adjacent, for ready reference and test. Instead, shafting has to be erected in bearings suspended from roofs, or supported on beams or carried against walls or in wall boxes, the datum faces for which are not in any degree even approximately accurate. Hence the methods of test used lie outside these, and consist chiefly of straining lines, plumb lines, and spirit-levels, supplemented with templets of special forms.

Alignments. The alignments of shafts must be correct in two directions—the horizontal and the vertical. Departures from these, even in slight degrees increase friction, heating up, power consumed, and scoring of brasses, so that realigning and insertions of packings are necessary after lengthy intervals. And since walls settle, and roofs expand or become distorted, and loaded beams yield, provision is often made in the bearings for effecting minute readjustments in both vertical and horizontal directions. Alignment is assisted by the employment of swivelling bearings, but the necessity for accurate erection remains. This work is more readily accomplished in a new building than in one which is largely occupied with machines, and with some existing shafting. The reason is, that the unoccupied floor can be utilized for the laying down of centre lines and for levelling—data from which the shaft centres can be obtained. It is not safe to seek data directly from the level of floors, or the faces of walls, or the flanges of girders. These faces are utilized, but only indirectly—not for aligning purposes.

Lining Out on a Floor. A clean and level floor is utilized for marking in horizontal plan the centres of a line or lines of shafting, determined in reference to the machine tools or countershafts which it has to drive, and to the engines or motors that drive it. A chalked line is used r this work, identical with that which a car-enter employs to snap the line of an extended w cut on timber. It is strained taut by a man each end, lifted at the centre, and released snap down sharply, when it leaves a fine halked line perfectly straight on the floor. A ail is driven in at each end for reference. arallel lines for short countershafts are snapped djacent at distances set by measurement. cross the longitudinal line, transverse lines re measured and marked to correspond with ne centres of the bearings to be bolted plumb ith the centres on the floor, the intersections of ne lines being marked with a nail.

Bearings Bolted under Beams. These are angers or hangdown bearings, the feet of which re bolted to the lower flanges of beams. A ooden templet is prepared to the same outlines s the foot, having a small centre hole corre-ponding with the centre of the bearing, and ith the intersection centre of the lines on the oor, and also holes bored in the same positions s those of the bolt holes in the foot of the earing. The bearing centre is plumbed with cord passed through the centre hole, and when he bob comes to rest on the intersection line n the floor the bolt holes are marked through, be bored with an auger if the beams are of imber, or drilled, if of steel. The bearings are ow bolted up only temporarily, because after olting there may a slight variation from a truly lumb position, and vertical adjustments will ave to be effected to line the shaft in the orizontal direction.

Corrective Alignments. The straining line is sed for ascertaining or correcting the lateral lignment of the bearings after they have een bolted up. The line is a length of fine ire pulled taut and laid alongside one edge of ne bearings, if they are all of the same bore. of different sizes, they are bridged, each with strip of wood on which the centre is scribed, nd the line is strained to check the centres. further check is to place a piece of shafting emporarily into the bearings in succession, and lumb to the line along the floor. The line may ang fron one side, Fig. 50, and half the shaft's iameter be added to the location of the bob, or may be looped over to fall centrally, Fig. 51.

To correct the alignment in the vertical irection, three methods are open to choice, ome one of which may be more suitable than ne others in different conditions. The spirit-vel is employed in conjunction with a piece of haft laid in the bearings in succession. A short level should not be relied on to test the truth of a long shaft, but a parallel straight-edge should be interposed, or a level permanently mounted on a straight-edge. And the level may be turned round end for end, to average its own readings. If successive lengths of shafts are checked thus, and the bearings adjusted vertically, the entire length will be in horizontal alignment.

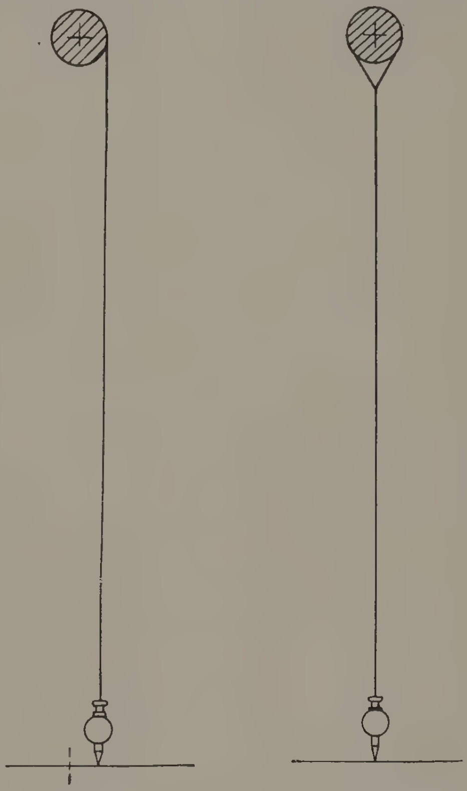


Fig. 50 Fig. 51

Plumbing from a Shaft to a Line on a Floor

If a suitable piece of shaft is not handy, or if the work is obstructed from any reason, then a piece of parallel rod may be laid in the bearings and two wooden suspension rods hung from it, shouldered at the bottom parallel with the top to receive a strip supporting the level, or to which the level is permanently fastened. The same method can be employed with shafting in place, Fig. 52. Another method is to use a theodolite set to take a sight adjacent to each bearing. A strip of wood is hung from the shaft having a line scratched across it. The level is sighted to this mark, and the strip is moved in succession and hung in the vicinity of each bearing, and sights are taken until all the bearings have been adjusted. Vertical corrections are made with wood packings either directly, or between the foot and a sole plate. These packings are also desirable to lessen the jar and noise that may result when metal is bolted directly to metal.

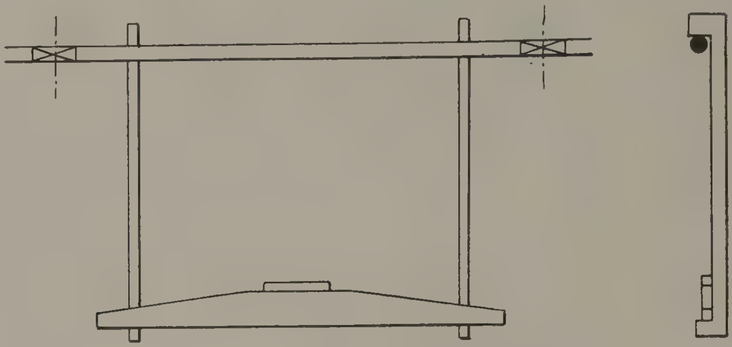


FIG. 52. LEVELLING SHAFTING IN PLACE

Bearings Attached to Walls. When bearing brackets are bolted to walls, the datum line on the floor is employed to obtain the lateral positions of the centres of shafts, the wall being unreliable. The height of the shaft and its vertical alignment are determined and corrected by the methods just described, working from one or both ends. An alternative way for vertical aligning when the floor is clear of machines, is to nail strips of board on the floor below the location of each bearing, and correct the thicknesses with a parallel straight-edge laid from one to another, and a spirit-level until they are all reduced to one height. Then on these the lateral centre line is marked with the straining line, and lines drawn across this give by their intersections the centres of the bearings.

From the faces of these, direct measurement can be taken to the bearing brackets when these are being adjusted in their positions. As they have feet for bolting to the wall, a templet is cut to the outline of the foot. On it, vertical and horizontal lines are scribed giving the shaft centre, and holes are bored to correspond with the bolt holes, Fig. 53. The templet is laid against the wall by measurement from the datum lines, and the bolt holes are marked through it on the wall, as in Fig. 53. The vertical centre lines are obtained by means of a chalked plumb line, snapped at each location, in the manner shown at the ends of Fig. 53. If the bearings stand out to a considerable distance from the wall, a frame is made with a foot to lay against the wall, and the distance out of the bearings is marked approximately, but set exactly with a plumb bob on the centre of the floor strips.

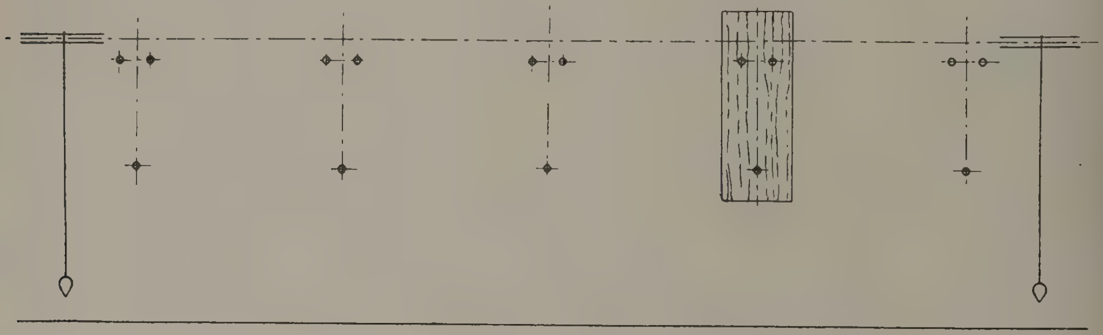


FIG. 53. TEMPLET USED TO MARK BOLT HOLES

Shafts at Right Angles. Cases arise in which a ne of shafting has to be set at right angles ith the longitudinal lines, and also where afting has to be run in parallel lines on opposite des of a partition wall, which prevents taking irect measurement. In these cases the geo- etrical construction, 3, 4, and 5 is employed, hich, when a square cannot be used, yields a right angle. Parallel lines on opposite sides of a wall will be obtained with two such constructions, Fig. 54, one on each side. One is made of sufficient length to carry the line through the doorway, and another prolongs it. Shafts that lie back near the wall can be measured off, and bearings erected therefrom by the methods already described.

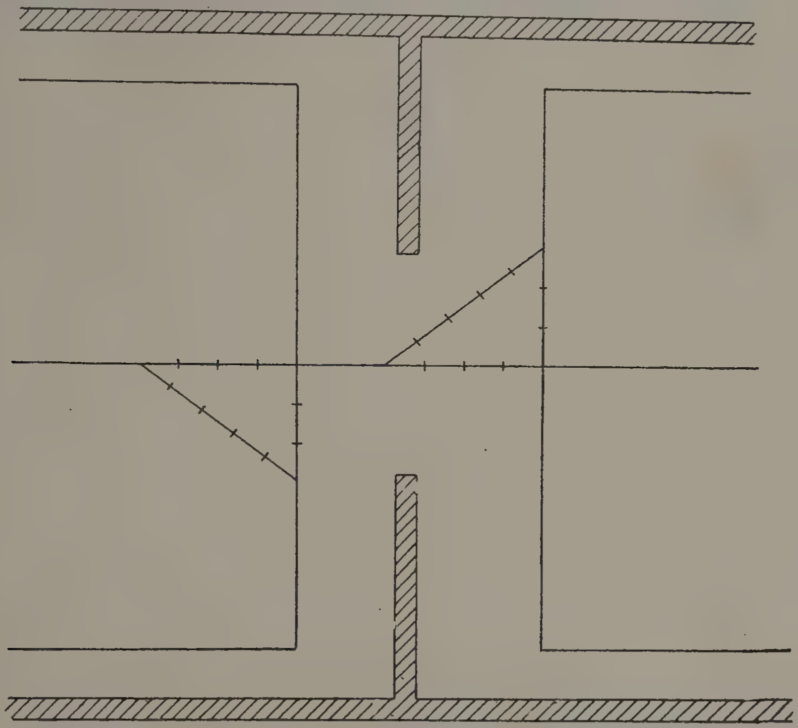


FIG. 54. OBTAINING PARALLEL LINES ON OPPOSITE SIDES OF A PARTITION WALL

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.Inst.N.A.

LESSON I

INTRODUCTION

The steam turbine is at once one of the oldest and one of the most modern forms of prime mover. We are told that Hero constructed a small reaction turbine, driven by steam, at Alexandria, in the year 120 B.C. Hundreds of years after, in the year A.D. 1628, we find that Branca arranged a steam jet to play on the buckets or blades of a small wheel, and thus constructed the first impulse turbine.

Engineers of to-day are inclined to look back upon these small beginnings with a certain amount of good humoured contempt, but there is no denying the fact that these crude inventions were the forerunners of what is probably the most important heat engine of our time. It required the ceaseless application of mechanical science and art, largely developed during the evolution of the reciprocating steam-engine to bring the steam turbine to its present exalted position. Amongst the many engineers who have helped to develop and improve the modern steam turbine from about 1880 to the present time one might mention the well-known names of Sir Charles Parsons, Gustav de Laval, Prof. Rateau, Curtis, Zöelly, and the brothers Ljungström.

Advantages of the Steam Turbine. The comparatively rapid development of the steam turbine has been no doubt due to the many advantages possessed by this engine over other types. The more important of these may now be enumerated as follows—

(*a*) In all other engines the propelling force is due to the static pressure of the working substance, be it steam or gas. This static pressure must, of necessity, be variable and hence the turning or twisting effect on the shaft for each cylinder is not uniform in amount. It is true that the turning moment may be made more uniform by a suitable arrangement of the cylinders, but even then some variation usually occurs. In the steam turbine, the energy due to static pressure is first converted into kinetic energy by causing the steam to issue as a well-formed jet after expanding in a *nozzle*. The jet of steam is then directed on to curved *blades* attached to the rotor, and due to the change of momentum which occurs when the steam particles are deflected by the blades into a new course, there is a dynamic force applied to the blades. Such a force is constant, and hence the twisting effect on the shaft is quite uniform and the speed does not, therefore, tend to fluctuate. For this reason, the steam turbine is admirably suited for driving electric generators, centrifugal pumps, fans, blowers, and similar machines.

(*b*) The propelling force is applied directly to the rotating element and not by a crank and connecting rod mechanism, as in reciprocating engines.

(*c*) Owing to the fact that the rotor is a symmetrical and inherently balanced body, the turbine is eminently suitable for high speeds of rotation. In practice, the turbine rotors are never *perfectly* balanced, but means are now available whereby the out-of-balance forces can be made a very small quantity.

(*d*) The steam consumption of the modern steam turbine is lower than that of the best steam engine. The amount of work done per lb. of steam in large turbines of good design exceeds 80 per cent of that theoretically possible in a perfect heat engine operating on the same cycle between the same limits of pressure and temperature.

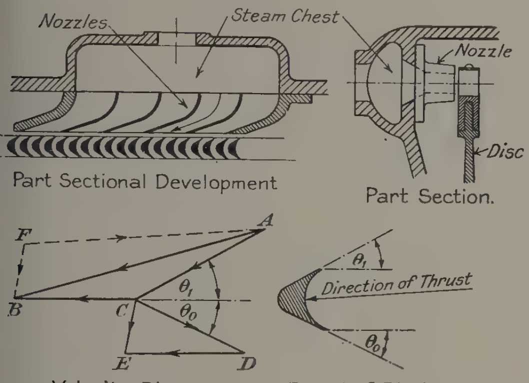
(*e*) A feature of considerable practical value is the entire absence of internal lubrication in steam turbines. Two important advantages follow from this. In the first place, the exhaust steam is not contaminated with oil vapour and the condensate may be fed into the boilers without passing through filters. Secondly, there is a considerable saving of lubricating oil.

(*f*) As the only parts of the turbine which require lubrication are the bearings supporting the rotor spindle, governor shaft bearings, and one or two gear wheels, and as there are no sliding parts, the amount of wear is negligible and the cost of lubricating oil is almost a negligible quantity.

(*g*) The comparative ease with which steam turbines may be designed and constructed for large powers (up to 50,000 kw. in a single casing) makes it an ideal prime mover for arge power stations and for the propulsion of ig ships.

(*h*) As will be shown later, the steam turbine s capable of utilizing a high vacuum to much etter advantage than the reciprocating steam-ngine.

Fundamental Principle of Action of Steam urbine. Every steam turbine consists, essenially, of one or more stages in which are arranged ozzles and blading. The function of the ozzles is to convert the initial pressure energy f the steam into kinetic energy and to form a roperly shaped jet. The jet of steam so ormed is then directed on to the moving blades f the turbine and, due to the curvature of the lades, the jet is deflected through a certain ngle. This deflection causes a change of omentum of the steam and, since *force* must e applied to the steam by the blades in order o effect this change of momentum, an equal and pposite force must be applied by the steam to he blades, and it is this force which is available or performing work on the turbine blades.



Part Sectional Development

Part Section.

Velocity Diagram.

Detail of Blade.

FIG. 1. DIAGRAM ILLUSTRATING THE PRINCIPLE OF THE IMPULSE STEAM TURBINE

The principle as briefly outlined above may e further illustrated by Fig. 1, which shows iagrammatically the nozzles and blades of a imple impulse turbine. A diagram of velocities s also shown, in which the vector AB represents he initial velocity of the steam as it leaves the ozzles. The velocity of the blades is repreented by CB, and hence the velocity of the team relative to the blades at inlet must be epresented by AC, the direction of flow being t an angle θ_1 to the plane of the turbine wheel.

A section through a typical impulse turbine lade is shown to the right of Fig. 1. In order hat the steam may flow into the channels between the blades without shock, the flat surface on the leading side of the blade must also be arranged at an angle equal to or slightly greater than θ_1. The direction of flow of the steam at outlet from the blades is decided mainly by the angle θ_0 of the blade. The velocity CD will be somewhat lower than the relative velocity AC, owing to friction and shock losses in the blade channel. It has to be remembered, however, that CD is a relative velocity. If we now draw DE equal to the velocity of the blades, we obtain CE, the final *absolute* velocity, or velocity relative to the casing of the turbine.

We have now the initial *absolute* velocity AB, and also the final value CE. To find what change of velocity must have been produced, we draw FB parallel to and equal to CE. It follows that FB must be the *resultant* of the *component velocities* FA and AB.

Vectorially, $FB = FA + AB$.

Therefore, FA must represent the velocity change produced by the curvature of the blades. From Newton's second law of motion we know that

Force (in poundals)

$$= \frac{\text{mass} \times \text{change in velocity}}{\text{time}}$$

Force (in poundals)

$$= \frac{\text{mass}}{\text{time}} \times \text{change in velocity}$$

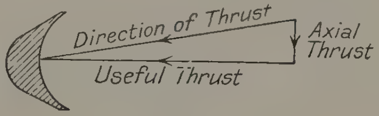


FIG. 2

If F = force in lb. weight steadily applied by the steam to the blades.
,, w = mass of steam per second.
,, g = acceleration of gravity (32·2 ft./sec.^{2}).

Then in the example shown in Fig. 1, the thrust of the steam on the blades is equal to

$$\frac{w.AF}{g} \text{ lb. weight.}$$

If the resultant thrust is inclined to the direction of movement of the blades, it may be resolved into two components, as shown in Fig. 2, giving a useful tangential thrust and an ineffective axial thrust.

FANS AND BLOWERS
THEIR CONSTRUCTION AND OPERATION

By F. G. Whipp, A.M.I.Mech.E., A.M.I.H.V.E.

LESSON IV

ENERGY IN AIR STREAM

The work done in giving motion to a current of air, independent of that absorbed by friction in the machine which creates that motion, is known as the *air horse-power*, and is usually symbolized a.h.p. A current of air possesses a certain *aero-motive force* dependent upon its density and velocity.

Force is the cause which produces motion or tends to produce motion, and in the case of a fan is supplied by the revolving wheel. In order that work be performed, force must be exerted through space, and if a force of 1 lb. weight be exerted through a foot of space, then a ft.-lb. of work has been done and a ft.-lb. of energy expended.

When a weight is able to fall it possesses a potential energy, or in other words, is capable of performing work, and the amount of that work will be equal to its weight in pounds multiplied by the difference in levels in feet to which it can fall, assuming there is no friction or other impediment to its free motion.

When a body is in motion, it possesses *kinetic energy*, equal to half its *mass* multiplied by its *velocity squared*. The *mass* of a body = its *weight* divided by *gravity*, so that kinetic energy

$$= \frac{w \times v^2}{2g} \text{ ft.-lb.}$$

Where w = the weight of the body in motion.
v = its velocity in feet per second.
g = 32·2 ft. per second per second.

Expressed in terms of cubic feet of air we get—

$$\frac{Qdv^2}{2g} \text{ ft.-lb.}$$

Where Q = Quantity of air in cubic feet per second.
d = density of air at temp. T and barometric pressure p.

1 h.p. is equivalent to 550 ft.-lb. per second. so that

$$\text{a.h.p.} = \frac{Qdv^2}{2g \cdot 550}$$

Density. From Boyle's law we know that the volume of a gas varies inversely as the pressure and according to Charles' law directly as the absolute temperature. It is sufficiently accurate to assume a constant barometric pressure of 30 in. of mercury for normal fan calculations, and in the majority of cases met with in propeller fan application an average temperature of 50° F. can usually be assumed. These conditions give a density of ·0781 lb. of air per cubic foot which simplifies the a.h.p. formula to

$$\frac{Q \,.\, \times \cdot 0781 \times v^2}{64 \cdot 4 \times 550} = \cdot 00000223\, Qv^2$$

Where temperatures other than normal are met with, as for instance in refrigerating, drying, induced draught plants, etc., the value of d must be correctly written into the formula as the power will be influenced considerably.

To simplify the calculations, a table is given below which indicates the value of d at various temperatures.

We will now apply the formula given in the foregoing to an actual case, and determine the power of a 30 in. diameter fan delivering 5,000 cub. ft. of free air per minute at normal barometric pressure and at 32° F. Diameter of central boss being $7\frac{1}{2}$ in.

The total area of fan

$$= \pi R^2 = 3 \cdot 14 \times 1 \cdot 25^2 = 4 \cdot 902$$

Weight of 1 cub. ft. of Dry Air at T° F., 30 in. Barometric Pressure

Temp. ° F.	10	15	20	25	30	32	35	40	45	50
d	·0847	·0838	·0829	·0821	·0813	·0809	·0804	·0796	·0788	·0781
Temp. ° F.	55	60	65	70	75	80	85	90	95	100
d	·0773	·0766	·0759	·0751	·0744	·0737	·0731	·0724	·0717	·0711

The area of the central boss

$$= \pi r^2 = 3{\cdot}14 \times {\cdot}3125^2 = \underline{{\cdot}306}$$

$$\therefore \text{Effective area of fan} = 4{\cdot}596 \text{ sq. ft.}$$

From which we obtain the velocity of the air ream passing through the fan as

$$\frac{5000}{60 \times 4{\cdot}596} = 18{\cdot}13 \text{ ft. per second.}$$

$$\therefore \text{a.h.p.} = \frac{5000 \times {\cdot}0809 \times 18{\cdot}13^2}{60 \times 550 \times 64{\cdot}4} = {\cdot}0625$$

Efficiencies. The efficiency of any machine : system is the measure or ratio of useful ork done to the total energy dissipated by ie apparatus in its efforts to perform that ork, or it may be defined as the ratio of the ieoretical ideal against practical working onditions.

In fan work there are three efficiencies to be ckoned with which are *Mechanical*, *Volu-etric*, and *Pressure*. The first two only concern s in the treatment of volume fans.

Mechanical Efficiency. We have seen how ie a.h.p. is calculated and, from an exam-ation of its derivation, it will be easy to e that it represents an ideal figure and does fact take the place of i.h.p. in the case of an igine. The energy dissipated in overcoming ie friction of the fan spindle in its bearings, elt friction and even the friction of the air iring its course through the bladed wheel ust all be added to this figure in order to find ie total work done in moving the air.

Mechanical efficiency therefore equals

$$\frac{\text{a.h.p.}}{\text{b.h.p.}} \quad \frac{\text{(air horse-power)}}{\text{(brake horse-power)}}$$

From a purely engineering point of view this ctor is of great importance, but from the fan nstructors' viewpoint, *volumetric efficiency* mes first.

This is a measure of the ratio of the actual iantity of air delivered by a fan against the iantity that would be delivered if there was slip.

Unfortunately, the want of standard methods calculation in the fan industry leaves much be desired, and the method of determining the lumetric efficiency does not appear to be uni-rsally agreed upon, but its thorough under-inding is of such importance to the fan engineer at it will be discussed at some length. In Fig. is given a set of characteristic curves, which is e mean result of a series of tests on various ikes of 30 in. volume fans. Naturally, the behaviour of the different fans was not all uniform, so that average curves have been constructed which, nevertheless, present a very fair idea of the results that may be expected from any of the well-known makes of fans of similar dimensions.

Observing first the curve marked *revolutions and capacity*, it will be seen that it is not perfectly straight, which it should be if the revolutions and volume were in direct proportion to each other. The deviation, however, is not very pronounced, so that calculations on the direct proportional basis will do for practical purposes, and for any given fan the volume will vary as the speed.

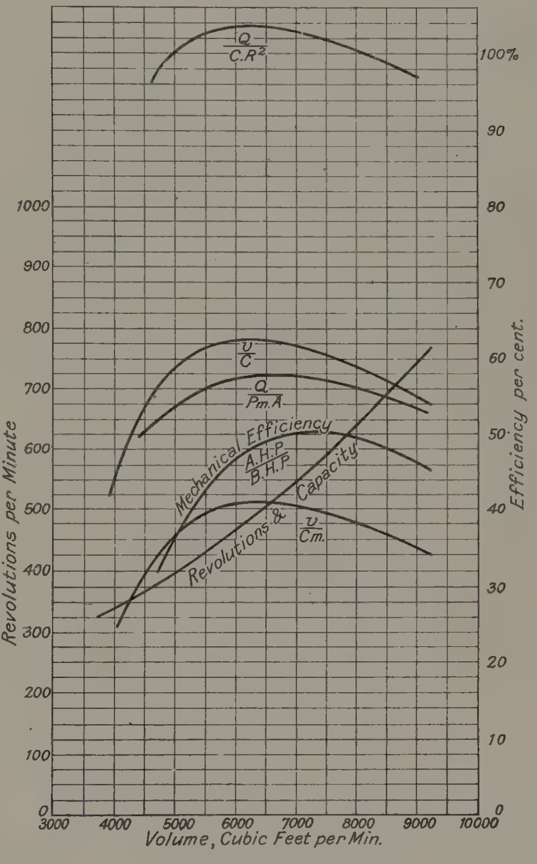


FIG. 10. CHARACTERISTIC CURVES OF 30 IN. DIAMETER VOLUME FAN

(Free Air)

Speed Range. The curve also shows us that the speed range for useful work is limited, as the curve will flatten at the lower end and rise more abruptly at the top, which means that a considerable speed must be maintained before any practical volume can be moved at all, and that above a certain maximum speed the air volume will cease to increase appreciably.

The abscissa of the graph represents the volume, and if we follow the ordinate at 5,000 cub. ft. per minute, we shall see that the 30 in. fan taken previously for our working example would have to run at 400 rev. per minute for the duty specified. Continuing up this line until the mechanical efficiency curve is reached a reading of 36·25 per cent is recorded, so that the b.h.p. absorbed by the fan will be ·0625 ÷ ·3625 = ·172.

Volumetric Efficiency. The diversity of opinion among authorities on fan work, as to how this factor should be arrived at, makes it somewhat difficult to compare the performances of different fans without a knowledge of the particular formula employed in each case for calculating the volumetric efficiency.

As a matter of fact, there is a standardization committee in being at the present time, who are investigating the whole problem with a view to setting a standard system of calculations.

The following are three equations, all of which are, or have been, used to determine the volumetric efficiency—

$$\frac{Q}{CR^2}, \quad \frac{V}{C}, \quad \frac{V}{C_m}$$

Where Q = volume in cubic feet per second.
C = circumferential velocity of wheel in feet per second.
C_m = average circumferential velocity of wheel in feet per second.
R = radius of wheel in feet.
V = average velocity of air in feet per second.
P = pitch of blades in feet.
A = area of fan wheel in square feet.
N = number of revolutions per second.
P_m = mean pitch velocity in feet per second = $C_m \cdot \tan \alpha$.

Comparisons of a given design of fan may be made with either of the above equations, but in no case will a correct conception be gained if they be employed on fans of different design, which, after all, is the very reason that volumetric efficiency is of so much importance.

Considering the first example, $\frac{Q}{CR^2}$, if the denominator be investigated it will be found to be equivalent to $A \times$ diameter $\times N$, which does not take into account the blade angle at all, and the same result would be obtained no matter what this angle was. It would, however, be applicable to a fan in which the pitch of the blades (assumed helical) is made equal to the outside diameter of the blades. There are other considerations, however, to be reckoned with as we shall see. Equation 2 is a comparison of the air speed and the peripheral speed of the fan wheel, and is largely adopted in the fan trade. As we have seen, the speed of the air stream is not constant over the whole of its cross-section, so that this formula will not give a true indication of a fan's performance.

The third equation has the mean circular velocity for its denominator, and this appears to be a much fairer method, but still the blade pitch is ignored.

If we define volumetric efficiency as: *The measure of the volume actually delivered against the volume that would be delivered by the fan if there were no slip and if the fan was truly helical in action,* then the following equation will be correct—

$$\frac{Q}{P_m \cdot A}$$

As an example of applying this formula, let us again take the 30 in. fan with helical blades delivering 5,000 cub. ft. of air per minute when running at a speed of 400 r.p.m., and assume a blade angle of 46° at $9\frac{3}{8}$ in. radius. What will be the volumetric efficiency?

$$C_m = \frac{\text{root velocity of blades} + \text{tip velocity}}{2}$$

$$= \left[\frac{(\cdot 625 \cdot \pi \cdot 400) + (2{\cdot}5 \cdot \pi \cdot 400)}{2}\right]$$

$\div\, 60 = 32{\cdot}708$ ft. per sec.

$\tan 46° = 1{\cdot}0355$

$\therefore$ mean pitch velocity $= P_m =$ $32{\cdot}708 \times 1{\cdot}0355 = 33{\cdot}86$ ft. per sec.

and volumetric efficiency per cent

$$= \frac{5000 \times 100}{60 \times 4{\cdot}596 \times 33{\cdot}86} = 53{\cdot}54$$

and air volume $= \dfrac{A \times P_m \times \text{vol. eff.}\%}{100}$

= cub. ft. per sec.

and revolutions per second

$$= \frac{Q \times 100}{\text{eff.}\% \times A \times D_m \times \tan \alpha \times \pi}$$

Where D_m = mean diameter of blades.
and α = blade angle at mean diameter.

STEAM CONDENSING PLANT

By John Evans, M.Eng.

LESSON VI

AIR PUMPS AND AIR EJECTORS

Air Leakage. The most vital part of any high vacuum condensing plant is the air pump, as the successful operation of the plant depends upon the efficiency with which air and incondensable gases are removed from the condenser. Progress in this branch has accordingly been rapid. The actual amount of air leakage into the system is very difficult to determine, and depends, not only upon the weight of steam condensed, but also upon the general "*air-tightness*" of the plant. Until of recent years, the air pump capacity required for a condensing plant of given steam duty was left in the hands of the manufacturer, but it is now customary to allow for air leakage, according to rules given by the B.E.A.M.A. (British Electrical and Allied Manufacturers Association), which, by giving liberal air pump capacities, allow a sufficient margin to make the plant comfortable of operation.

B.E.A.M.A. Air Ratings.

Land Surface Condensers—

Let Q = weight of steam condensed in lb. per hour.

W = weight of air to be removed in lb. per hour.

For h.p. turbines $W = \frac{0{\cdot}5Q}{1000} + 3$.

,, mixed pressure and low pressure turbines

$$W = 1{\cdot}5\left(\frac{0{\cdot}5Q}{1000} + 3\right)$$

,, reciprocating engines

$$W = 2{\cdot}5\left(\frac{0{\cdot}5Q}{1000} + 3\right)$$

Jet Condensers.

For jet condensers, allowance must also be made for air entering with the injection water, this addition being at the rate of 15 lb. of air per 1,000 gals. of water per minute.

$$\therefore\ W = \frac{G.P.M.}{1000} \times 15 = \frac{G.P.H.}{4000} \text{ lb. per hour.}$$

In marine installations, steam-driven auxiliaries usually exhaust into the condenser, and the air leakage is slightly in excess of that determined by the given rules.

The capacity test for an air pump as specified by the B.E.A.M.A., is such that the pump must be able to maintain the required vacuum in a closed vessel, when handling the rated quantity of air, admitted to the vessel through a standard air nozzle, the air being saturated with water vapour at a temperature 5° F. above the cooling water inlet temperature.

Air Pump Capacity for Jet Condenser. A jet condenser, working with a turbine, condenses 20,000 lb. of steam per hour, and maintains a 28 in. vacuum when supplied with 1,250 gals. of injection water per minute at 70° F.

Air entering with steam + leakage

$$= \frac{0{\cdot}5 \times 20{,}000}{1000} + 3 = 13 \text{ lb. per hour.}$$

Air entering with injection water

$$= \frac{1250 \times 60}{4000} = 18{\cdot}75 \text{ lb. per hour.}$$

∴ The air pump must be capable of dealing with $13 + 18{\cdot}75 = 31{\cdot}75$ lb. of air per hour, saturated with water vapour at a temperature of $70 + 5 = 75°$ F.

Classification. Air pumps may be divided into three classes—

(*a*) Reciprocating,
(*b*) Rotary,
(*c*) Ejector,

their order indicating the lines on which they have been developed.

Reciprocating Air Pumps. The earliest form of air pump is the single-acting reciprocating type, Fig. 23, which deals with both air and condensate. It consists of a bucket *B* which reciprocates in a cylinder provided with a delivery plate *D*, *B* and *D* having flat circular india-rubber valves *V* and *W*. A foot valve *F* is provided between the pump and condenser. During the downstroke of the bucket *B*, the foot and delivery valves *F* and *W* remain closed, and the air and water are forced from the underside of the bucket through the bucket valves *V* to the top side. On the upstroke, the bucket valves *V* are closed by the weight of the water on them, and the air and water are finally discharged to the hotwell through the delivery valves *W*.

The Edwards air pump which followed the type described, was a distinct improvement, as it had no foot or bucket valves (Fig. 24). The clearance volume at the top of the stroke is comparatively small, giving this type of pump an increased volumetric efficiency. Ports are provided in the cylinder wall near the bottom, through which air and water pass to the top side of the plunger, both being discharged to atmosphere on the upstroke as in the previous case.

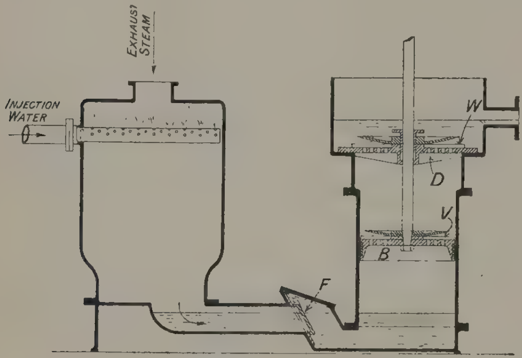


FIG. 23

Reciprocating pumps, while being suitable for reciprocator work where only a medium vacuum is required, are entirely unsuitable for turbines exhausting into a high vacuum. The increased specific volume of air at very low pressures, together with low volumetric efficiencies makes this type of pump large and cumbersome. In many installations, the single-barrel pump has given place to two separate pumps, or one pump containing two barrels. Under these conditions, the "wet" barrel removes the condensate, and the "dry" barrel removes the air which is cooled in the barrel by an independent water supply. The condensate temperature is thus maintained as near as possible to the vacuum temperature, while cooling of the air increases its density, allowing a greater quantity to be removed per stroke.

The usual allowances for Edwards air pump are—

Vacuum	*Air pump capacity per lb. of steam condensed*
28 in.	0·9 cub. ft.
27 in.	0·75 ,,
26 in.	0·6 ,,
25 in.	0·5 ,,

EXAMPLE. To find the diameter and stroke of a single-acting reciprocating pump which removes air and condensate from a surface condenser.

Steam condensed per hour = 20,000 lb., vacuum at pump suction = 28·25 in., temperature at pump suction = 90° F. The pump has a volumetric efficiency of 85 per cent, and makes 60 working strokes per minute. Vacuum corresponding to 90° F. = 28·58 in.

$$\text{B.E.A.M.A. air rating } (w) = \frac{0{\cdot}5 \times 20{,}000}{1000} + 3 = 13 \text{ lb. air per hour}$$

Total pressure of vapour and air at pump suction = 30 − 28·25 = 1·75 in. Hg.
Vapour pressure at ,, ,, = 30 − 28·58 = 1·42 in. ,,
∴ Air ,, ,, ,, ,, = 1·75 − 1·42 = 0·33 in. ,,

$$\text{Volume of air and associated vapour to be removed} = \frac{w.R.T.}{P} = \frac{13 \times 53{\cdot}2 \times (460 + 90)}{0{\cdot}33 \times 0{\cdot}491 \times 144} = 16{,}350 \text{ cub. ft. per hour}$$

$$= 272{\cdot}5 \text{ cub. ft. per minute}$$

$$\text{Volume of water to be removed} = \frac{20{,}000}{60 \times 62{\cdot}4} = 5{\cdot}35 \text{ cub. ft. per minute}$$

$$\therefore \text{Theoretical pump displacement} = 272{\cdot}5 + 5{\cdot}35 = 277{\cdot}85 \text{ ,, ,, ,,}$$

$$\text{Actual pump displacement} = \frac{277{\cdot}85}{{\cdot}85} = 327 \text{ ,, ,, ,,}$$

$$= \frac{327}{60} = 5{\cdot}45 \text{ cub. ft. per stroke.}$$

∴ Diameter pump barrel = 20 in. say.
Length of stroke = 30 in. ,,

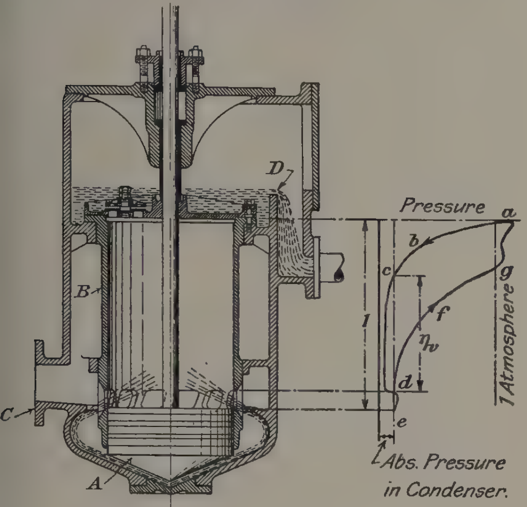


Fig. 24. Edwards Air Pump

Rotary Air Pumps. A section through a Iirrlees-Le Blanc rotary air pump is shown in ig. 25. It consists of a motor-driven rotating heel, which has a series of vanes fitted to the m. Arranged inside the wheel is a stationary ater box, fitted with a nozzle C so arranged that e rotating vanes, on passing the nozzle, shear off short columns of water E, imparting to them a high velocity. Pockets are formed between these columns of water, in which air and incondensable gases are trapped after being drawn through the pump suction branch. Both the water and air are then discharged into a converging cone, a high velocity being produced at the throat. In the diverging cone, the kinetic energy at the throat section is converted into pressure energy, allowing the air and water to be discharged against atmospheric pressure.

This type of pump condenses vapour associated with the air, and is thus its own air cooler and devaporizer. Although it overcomes some of the disadvantages of the reciprocating type, its introduction brought other difficulties, the most important being the wearing of parts due to water sheets being drawn over them at high velocities, and the difficulty of creating a vacuum rapidly when starting up.

The hydraulic vacuum pump (Fig. 26) removes air and incondensable gases from the condenser by means of water jets forced at high velocity from a revolving wheel. The operating water is delivered to the jet-transforming wheel by a pump, the vanes of the wheel being shaped so that the water gives the wheel a rotary motion. The water leaves the wheel in a number of

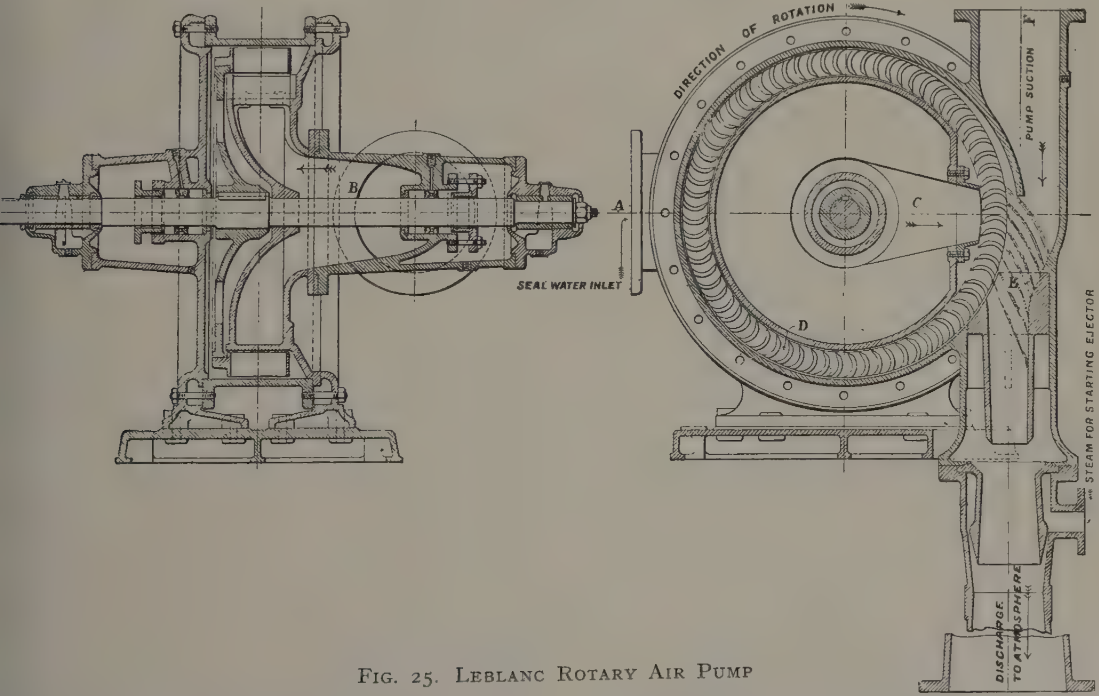


Fig. 25. Leblanc Rotary Air Pump

rectangular jets which entrain the air between them, and compress it in the diffuser.

Vacuum Augmenters. In the early days of steam turbine practice, it was discovered that the weight of the valves limited the highest vacuum obtainable with the reciprocating type of air pump. The pressure of air and vapour withdrawn from the condenser had to be sufficient to lift the foot and bucket valves, and after compression in the pump, to lift the delivery valves and pass through. Various attempts to remove this drawback were made, such as the adoption of light multiple disc valves, and the abolition of valves, as in the Edwards pump, but the first successful attempt was that of the Hon. C. A. Parsons, who introduced a steam jet and augmenter condenser to assist the air pump.

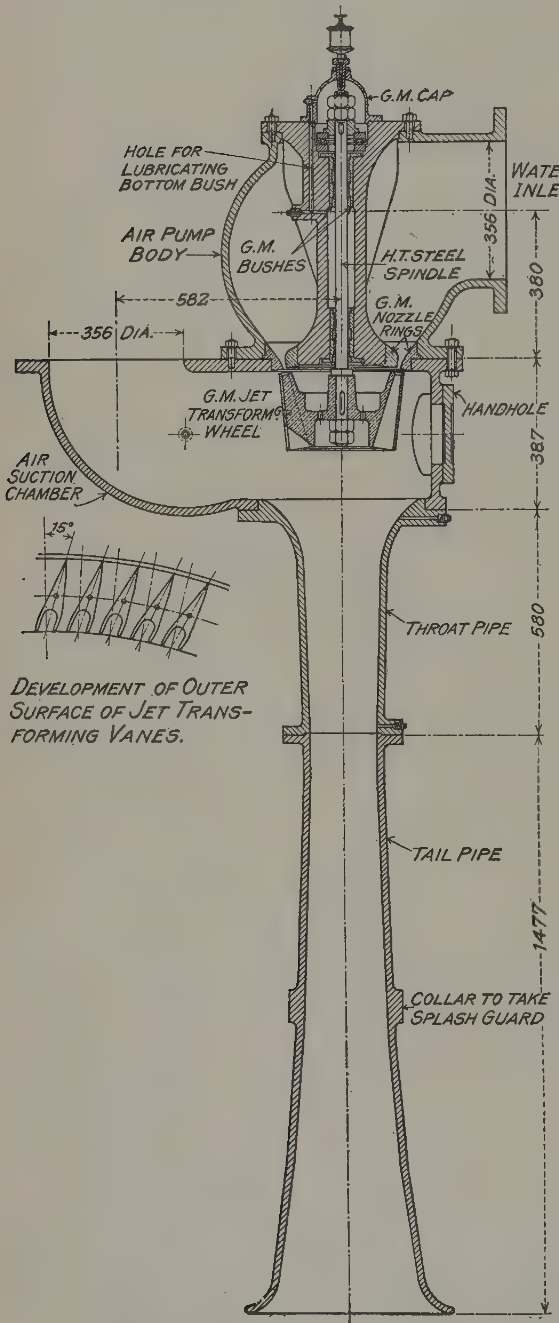


FIG. 26. HYDRAULIC VACUUM PUMP

The main condenser (Fig. 27) is provided with two suction branches, one connected to the reciprocating air pump which is of the "wet" and "dry" barrel type, and the other to the steam jet ejector *E*, which withdraws air and its associated vapour, and delivers it to the augmenter condenser *C*. The diffuser of the steam ejector compresses the air to within a few inches of the main vacuum, the condensed vapour and air from the augmenter condenser joining the main condensate in the pipe *P*. To provide a water seal and prevent the air partially compressed from re-entering the main condenser, the main condensate pipe is looped, the depth of the loop depending on the pressure difference between the main and augmenter condensers.

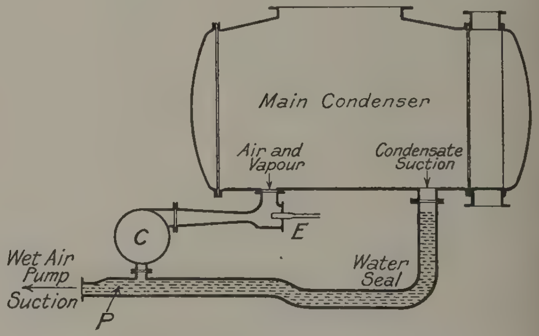


FIG. 27. PARSONS' VACUUM AUGMENTER

With this arrangement, the air handling capacity of the dry barrel of the reciprocating pump is increased, the air and vapour being first compressed in the augmenter, and the pump dealing with them at a comparatively low vacuum (approx. 26 in.).

Another successful type of reciprocating air pump with a steam jet vacuum augmenter is the Weir Paragon air pump, Fig. 28. It is usually

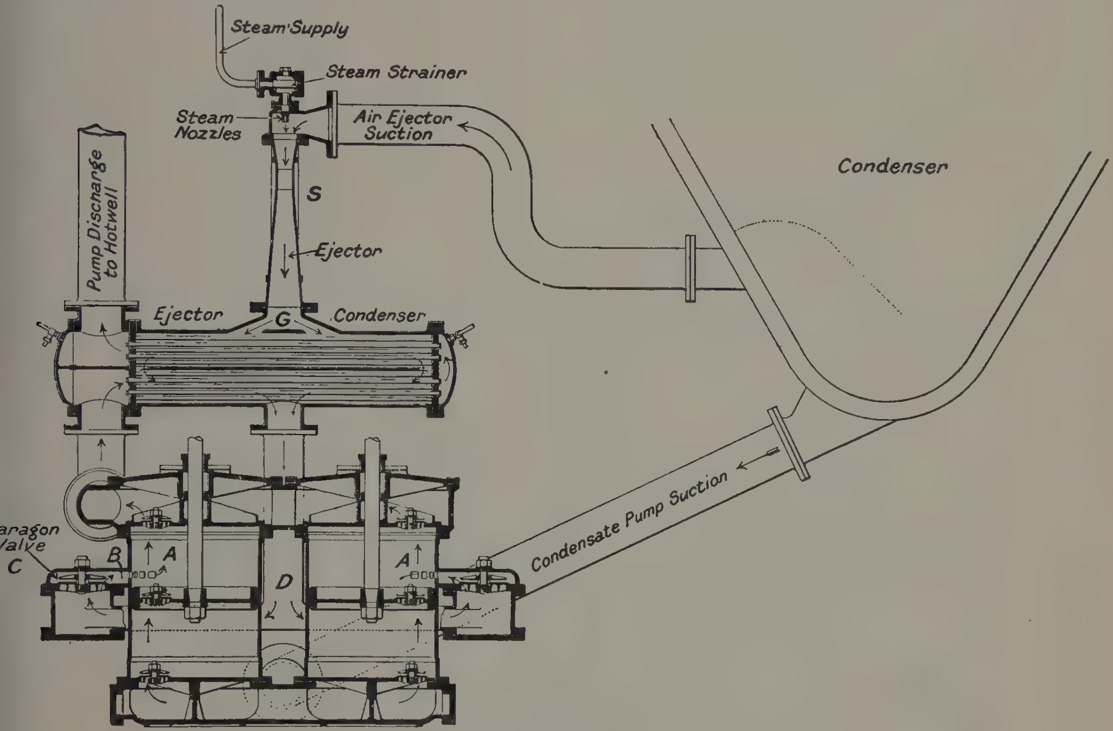


FIG. 28. WEIR'S PARAGON AIR PUMP

ised for marine purposes. The pump is steam-lriven, and of the twin-barrel type, both barrels naving foot, suction, and delivery valves, and ooth dealing with air and condensate. Arranged n the sides of the barrels are ports *A* in comnunication with the chambers *B*, which conain non-return valves *C*, similar in design to he pump valves. Air is withdrawn from the conenser by the steam jet ejector *S*, and delivered nto the augmenter condenser *G*, which is of the urface type. The steam discharged by the jector is condensed by the condensate disharged by the air pump, the heat in the augmenter discharge being conserved in the condensate. Air is withdrawn from the augmenter condenser outlet through the branch pipe *D*, which connects the two chambers *B*. When the bucket is at the top of its stroke, the ports *A* are in communication with the space under the bucket, and air flows into the barrel. On the down stroke, both the air and condensate are transferred to the top side of the bucket through the bucket valves, being finally discharged on the upstroke of the pump. The whole apparatus is compact and self-contained, being easily installed on board a ship.

NOTABLE EVENTS IN ENGINEERING HISTORY

By E. A. FORWARD, A.R.C.S., M.I.MECH.E.

LESSON VI

THE INTERNAL COMBUSTION ENGINE

THE changes in the pressure or volume of air when heated had been early noted, and were utilized as a means of performing work by the combustion of fuel in the hot-air engine. The earliest motor of this kind was designed by Sir George Cayley in 1807, while in 1816, Dr. R. Stirling introduced a more efficient type working with a closed cycle, compressing the air before heating and applying a regenerator. John Ericsson, in 1834, constructed an engine similar to Cayley's, while in 1852 he built some large engines for the propulsion of ships. Such engines, owing to their small temperature range, have a low efficiency, and are bulky for their power; they are now little used except for small powers where safety and simplicity are the chief requirements,

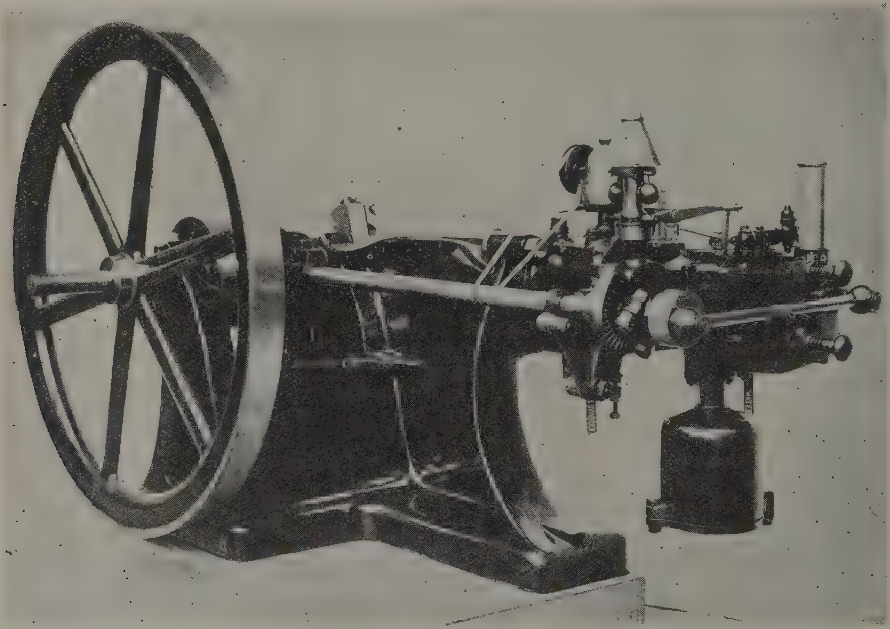


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FIG. 24. OTTO GAS ENGINE, 1876

By heating the air by exploding within it a gaseous mixture, the internal combustion engine was arrived at, which in the form of the gas or oil engine has been so generally adopted. In this, the heat obtained from the combustion of the fuel is utilized more directly than in the steam-engine, and with a higher thermal efficiency.

The earliest internal combustion engine was the gun, while as a means of obtaining mechanical power the explosion of gunpowder was tried in 1678-89 by Huygens and Papin. Attempts were made to construct a gas engine by R. Street, in 1794, and by S. Brown in 1823, but it was not until 1860 that J. J. E. Lenoir produced the first mechanically successful one (Fig. 23) facing p. 537; this worked with a cycle resembling that of a steam-engine, the ignition being electric. This was followed by Hugon's engine of 1865, with flame ignition, but both these engines had a heavy gas consumption, and vere soon superseded by the Otto and Langen tmospheric engine of 1866, which was the first commercially successful gas engine.

The greatest improvement in the internal ombustion engine was, however, the four-stroke ycle, practically introduced by Dr. N. A. Otto n 1876, when he compressed the explosive mixture in the working cylinder before igniting t (Fig. 24). By this system a much weaker mixture can be fired, giving a quieter explosion nd a more sustained pressure during the working troke, so that it has been adopted for most as and oil engines. Later improvements lay the direction of higher compression, the use f drop valves and the use of electric ignition place of the flame or hot tube.

While the Otto cycle was being developed, ttention was also given to producing an engine orking on a two-stroke cycle, notably by lerk, Robson, and Atkinson. The form now sed was patented by Sir Dugald Clerk in 1881, nd while it was unable to compete with the ur-stroke cycle in small sizes, it has been uccessfully adopted in large gas engines where has proved almost as efficient as the Otto type. The simpler valveless type of two-stroke engine, using crankcase compression, was invented by J. Day in 1891.

There has been a steady increase in the size and power of gas engines; in the larger examples two or more working cylinders are employed so as to secure a more uniform turning moment, while sometimes double-acting cylinders are used to obtain increased power from a cylinder of given size. These are often supplied with some cheap form of producer gas, or waste gas from blast furnaces.

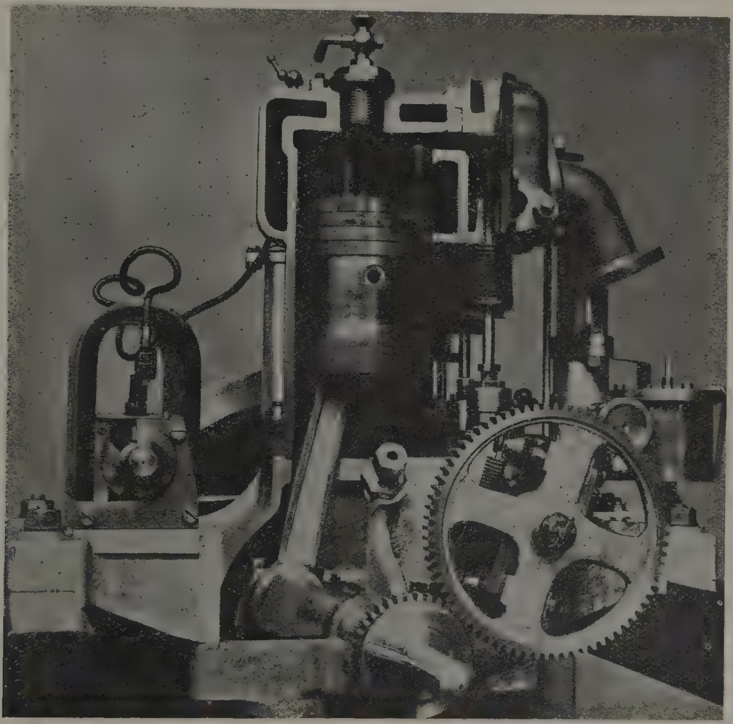

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FIG. 25. PETROL MOTOR, 1905

Oil and petrol engines work on the same cycles, but use liquid fuel, which is more easily stored and transported than is gas; provision has, however, to be made for spraying or vaporizing the fuel before being mixed with the air. The earliest petroleum engines, such as the Brayton engine of 1873, used light oil or spirit, which could be easily vaporized, but the risk attending the use of such fuels, and their greater cost, led to the more rapid development of the heavy oil type for stationary engine purposes. When, however, attention was turned to the application of the internal combustion engine to road carriages, the light oil motor was found most suitable, and a satisfactory form of light high speed engine was brought out by Daimler in 1884, and a more normal form by Benz in 1885. The invention of the spray carburetter by Maybach in 1893, and the Bosch magneto shortly afterwards, gave this motor the necessary reliability and easy means of regulation, while the use of two, four, six, and more cylinders ensured uniformity of turning moment. Most of these motors work on the four-stroke cycle (Fig. 25), but where a cheap engine is required, the Day valveless two-stroke type is much used, although less efficient.

The first practical oil engine using petroleum with a high flash-point was that introduced by Messrs. Priestman in 1885, who devised a special spraymaker and adopted electric ignition. In 1890, H. Akroyd Stuart patented an engine in which the fuel oil was sprayed into the air at the end of the compression stroke; ignition was effected by a hot bulb attached to the cylinder head, the heat of combustion being sufficient to maintain the temperature of the bulb after being heated at starting by an external lamp. In the ordinary form of Akroyd engine, however, introduced by Messrs. Hornsby shortly afterwards the oil was introduced into the cylinder during the admission stroke and compressed with the air into the hot bulb (Fig. 26). This, on account of its simplicity, became the first popular form of oil engine. Dr. R. Diesel, in 1893, independently developed the Akroyd cycle, but he carried the compression to such a high point that the temperature attained was sufficient to ignite the charge. Crude oils from almost any source can be used with this engine, and while it has a high thermal efficiency, its mechanical efficiency is lower, so that the brake efficiency is comparable with that of the best gas engines. The so-called semi-diesel engine, now so much used, operates on the Akroyd cycle, which is, moreover, specially suitable for two-stroke engines.

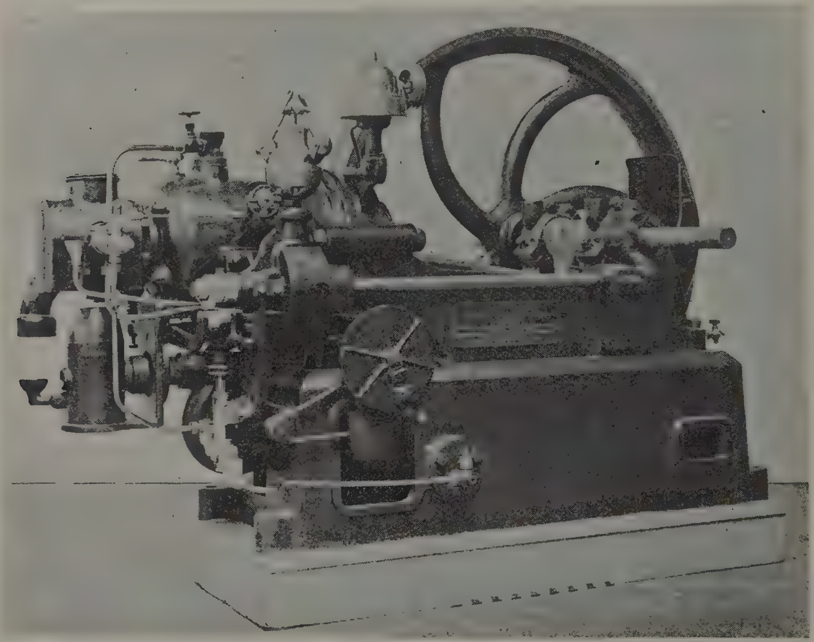

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FIG. 26. HORNSBY-AKROYD OIL ENGINE, 1893

The internal combustion engine has been applied to all purposes where power is required. The petrol motor made possible the enormous advance in mechanical road transport which has taken place during the past thirty years, aided, of course, by the provision of a plentiful supply of cheap fuel. It has also fulfilled the dream of centuries by making the flying machine practicable. Both light and heavy oil motors were early applied to marine propulsion, and the latter has now become a serious rival to the steam-engine under suitable conditions.

MACHINE TOOLS

By Joseph G. Horner, A.M I.Mech.E.

LESSON XI

BORING MACHINES

HE practice of boring has advanced in several rections. As a general statement, holes up about 3 in. diameter are drilled, and finished th reaming, those over that size are drilled cast) and finished with boring tools or by inding. All boring was done originally in the the, small and large alike. Cylinders were lted on the carriage of the slide rest, the upper ol equipment being removed, and fed past the ring bar with cutters, rotated between centres. ne method is still retained, though employed ss than formerly. Cylinders up to 6 ft. bore ve been mounted on the base-plate of a face the, and bored with a traversing head of tters, rotated between centres, and fed with a star wheel or differential gears. The large cylinders for the old reciprocating engines were generally bored with their axes vertically, the boring bar being revolved with worm gear, and fed. An early form of horizontal boring machine, the parent of more substantial designs, comprised a tee-slotted base, a boring bar adjustable in head, and tail standards, with provisions for speeding and feeding. Another transitional form for the smaller bores was that in which the boring spindle was stiff enough to perform the cutting without any support for the farther end. This, with radical alterations is now the most favoured for boring motor cylinders of from 3 to 4 in. diameter.

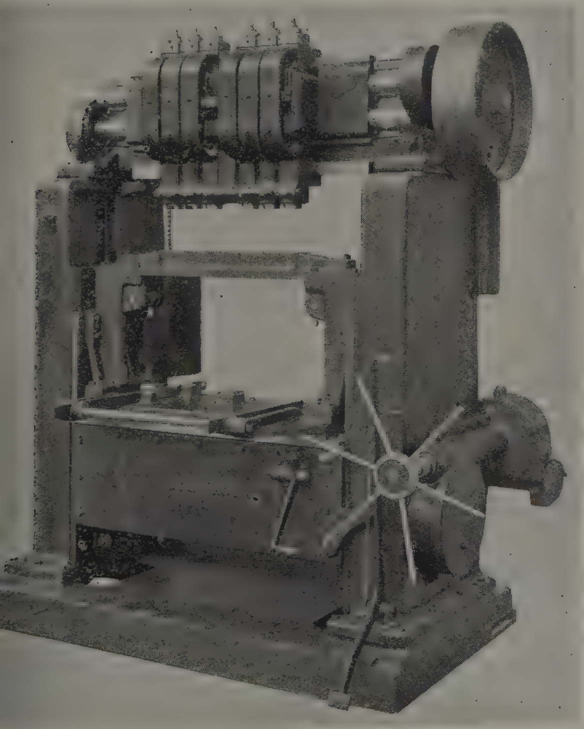


Fig. 59. Cylinder Borer with Jig

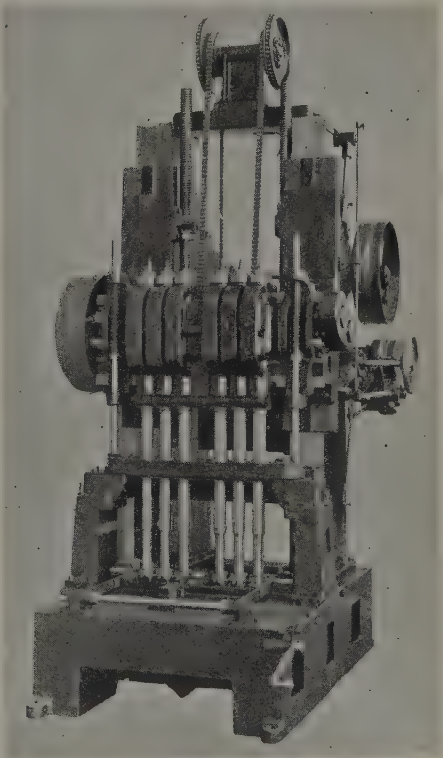


Fig. 60. Vertical Type Cylinder Borer with Jig

The standard types of boring machine at present in use may be grouped under three heads: vertical spindle machines, built on the model of the similar drilling machines, and used only for the smaller bores, horizontal spindle machines, very powerful for large bores, and usually including provisions for other functions, and the vertical boring and turning mills, that deal with the most massive articles.

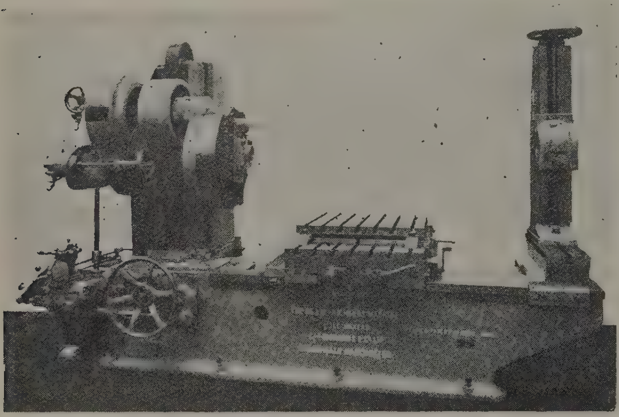


Fig. 61. Horizontal Boring Machine

Vertical Spindle Machines. These resemble in outlines the drilling machines of the same type, but they operate boring bars and heads of cutters instead of drills, and require more driving power. By far the larger number now built have multiple spindles, ranging from two to eight, and they are used chiefly for dealing with blocks of motor cylinders. These tell a story of evolution, for the early machines had but one spindle, like the drills, and the work was moved between gangs formed of separate units. Rough—and finish-boring would be done on adjacent machines, and reaming on a third, with perhaps an interval for seasoning—that is, to give time for the metal to accommodate itself to internal strains which might cause distortion after the removal of the hard surface skin. But now, as many spindles as are required are mounted on one head and rotated simultaneously, and are, except for highly repetitive service, adjustable for centres. The number of spindles is in a measure controlled by the sizes bored, as in the drills. Thus, there may be four 6 in., or six $4\frac{3}{4}$ in., or eight $3\frac{1}{2}$ in. spindles, and other sizes in similar proportions. The power required is large, from 15 to 20 h.p. belt, or motor drives are provided.

Two designs of these machines are built, one in which the spindles are strictly vertical, Figs. 59 and 60, the other, where they are inclined for convenience of handling deep cylinders, and those where boring bars are piloted. In Fig. 59 the cross beam that carries the spindle heads is fixed on top of the flanking columns, and the vertical feed is imparted to the work table. In Fig. 60 the feed is given to the beam that carries the heads. The heads are adjusted along the beam and clamped, and the spindles are driven with continuous spirals. These machines deal almost entirely with blocks of motor cylinders, which are frequently mounted in jigs, as shown in both figures.

Horizontal Spindle Machines. This group is used extensively for the heavier kinds of boring. But their functions are not restricted to that, for facing, milling, drilling, tapping, and screw-cutting are commonly included—these to avoid the time lost in transferring massive pieces between different machines, and resetting. Large numbers of these machines are built, having external resemblances, but with many variations in details. Their adaptabilities have increased in recent years. A single example only can be selected for illustration, one of the best known, the "Pearn-Richards," by George Richards & Co., Ltd., of Broadheath,

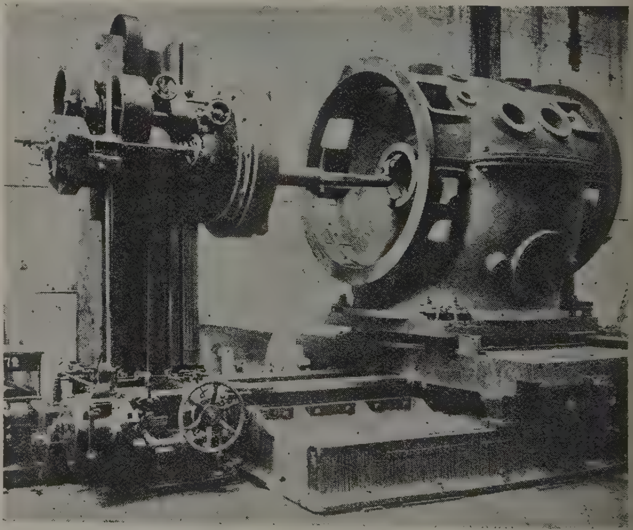


Fig. 62. Machine Boring Ljungström Turbine Casing

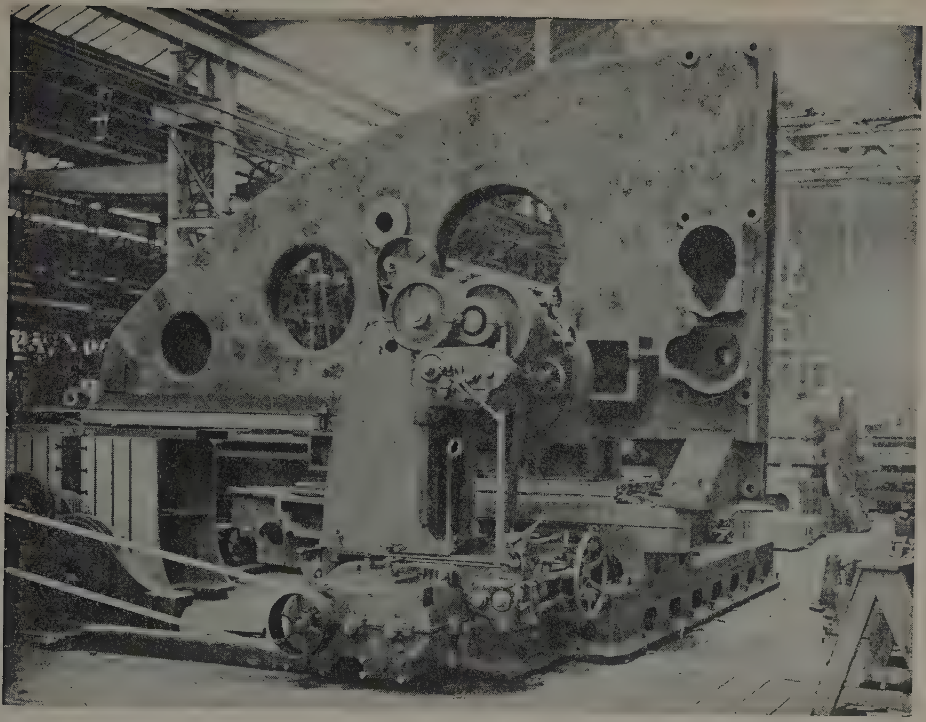


FIG. 63. FACING THE HOUSING OF A PLANING MACHINE

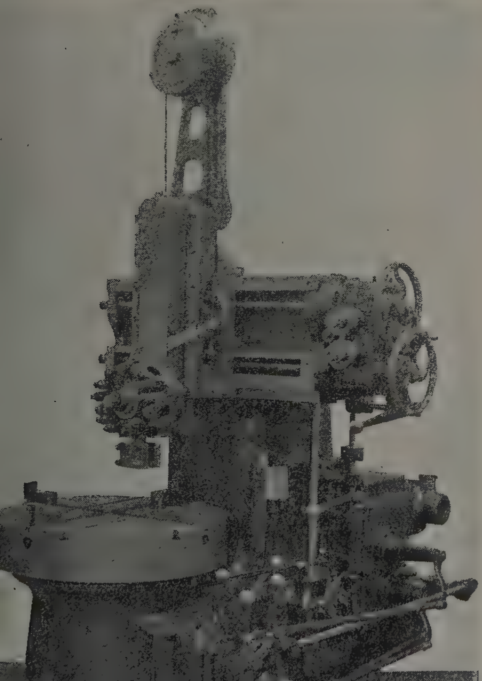


FIG. 64. BORING AND TURNING MILL
(Single Table)

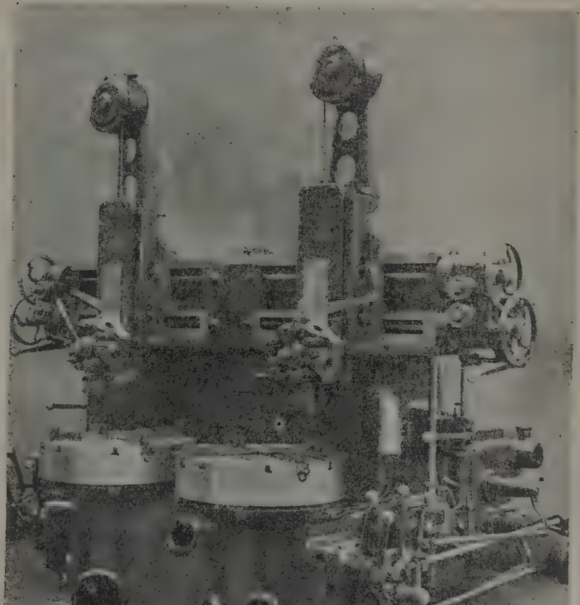


FIG. 65. DUPLEX BORING AND TURNING MILL

Manchester. It is illustrated by Fig. 61, and some specimens of work being done in subsequent Figs. 62 and 63. It is made in six sizes, which range from a facing capacity of 18 to 72 in. with lengths of bed and table traverses proportioned. That shown is a No. 3 machine, with a facing capacity of 36 in., and a distance of 6 ft. 7 in. between the socket for the boring bar, and the stay that supports its farther end. The spindle has a range of 32 speeds from 2·6 to 160 r.p.m., in geometrical progression through a gear box outside the bed, with levers. For work of large diameter the drive is direct to the facing head, so relieving the spindle of torsional stress. There are eight rates of feed to the table, giving from 100 to 8 cuts per inch, all capable of instant change and reverse. These feeds are applied to the longitudinal and transverse movements of the work tables, and to those of the vertical slide, and each is by power. The driving is by a single-belt pulley, or a constant speed motor, $7\frac{1}{2}$ h.p. being required for this machine.

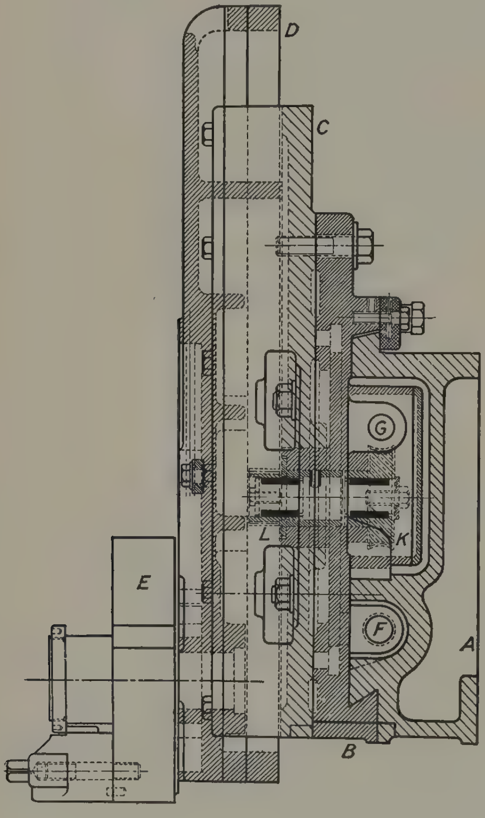


Fig. 66. Tool Slides of Boring Mill

For long bores, bars are used, slotted for cutters, driven with a taper socket in the head centred against a stop, supported in a bush in the boring stay, and carrying cutters for the simultaneous boring of bearings in engine beds. Bars range from $1\frac{1}{2}$ to 5 in. diameter. Snout boring bars—unsupported at the free end, straight or offset, the latter for cylinders larger in diameter than the diameters of the facing head—are used, the range of utility being increased with a telescopic fitting. Another attachment receives milling cutters. Facing tools are clamped in a slide that is traversed across the facing head with spur gears, the feeds of which are adjustable with a lever, roughing when fed inwards, and finishing outwards. Adjustments of $\frac{1}{1000}$ in. are read on an indexed boss, and a safety slipping device prevents overrunning of the travel. The table is compound, with a quartering stop for setting work at right angles.

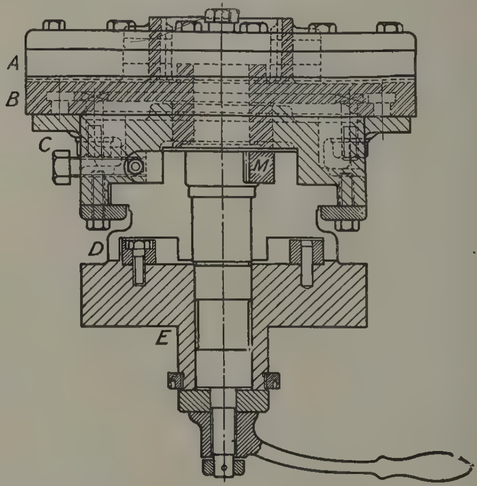


Fig. 67. Tool Slides-Transverse Section

Vertical Boring and Turning Mills. These may be regarded as face-lathes up-ended, bringing the face-plate horizontal instead of vertical. This is a great advantage in the work of chucking, and especially so when several segmental pieces have to be mounted for boring, facing, and turning. These have to be held and adjusted while being slung in the crane on the vertical face-plate until they have been centred, and packings and clamps adjusted and secured. No counterbalancing is necessary on the horizontal table which is often required when

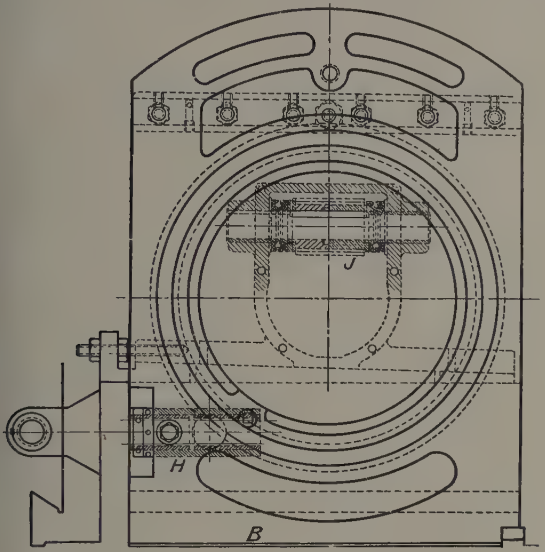


Fig. 68. Saddle of Tool Slide

pieces of irregular shapes, lop-sided, are rotated in the common lathe. The methods of gripping are like those on the common face-plates

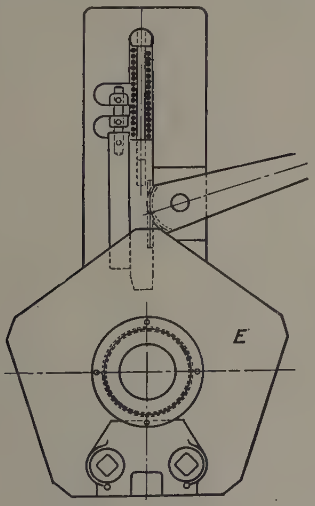


Fig. 69. Turret

of the lathes, comprising the clamping pieces, the usual forms of jaws, and special jaws, and the plates are tee-grooved. The spindles are massive, friction is reduced, and the spindle centred with a sloping form of bearing that is substituted for the Schiele curve used on the early machines. Continuous lubrication is provided for.

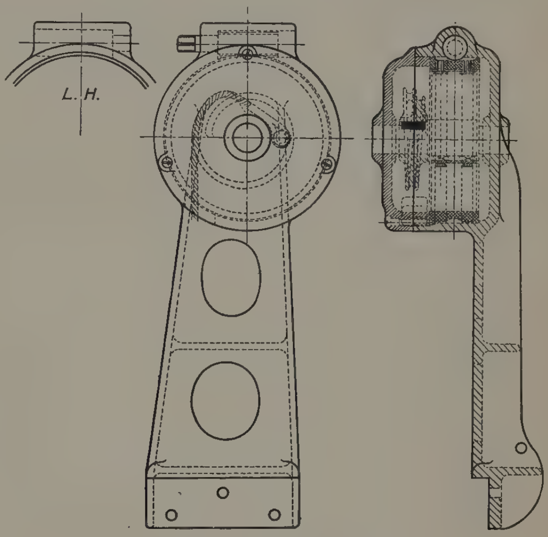


Fig. 70. Counterbalance

In the largest machines the table is supported besides on an annular bearing ring next the periphery. The driving is through a spur pinion and ring of teeth.

Figs. 64 and 65 show mills by Webster & Bennett, Ltd., of Coventry. Fig. 64 is a mill with a single table of 36 in. capacity, and Fig. 65 one with two tables (duplex type). These are made in capacities of from 20 to 48 in., and the tools are carried in turrets with five faces. The driving is done with a single pulley, or a motor, the changes being effected through gears enclosed, below the table to the right where the attendant stands, the handles being grouped there. The speed changes, twelve in number, are in geometrical progression, and are interlocked. The gears are chrome nickel steel forgings, heat-treated, and are splash lubricated.

The tool heads are carried on the cross slide, and are fed with twelve changes, vertically, horizontally, and angularly. They are operated with levers, are reversible, and interlocked, and have feed trips and safety devices. The slides have narrow guides adjusted with taper strips. The turret slides are counterbalanced with springs, which give a constant pull throughout the travel. The swivels are graduated in degrees for taper boring and turning. The turret faces, with $2\frac{1}{2}$ in. holes, receive boring bars with single or double cutters, drills and reamers, and tool holders for boring and turning tools of common shapes. The rotation of the turret permits of bringing five separate tools into successive action, and the use of duplex tables doubles production on repetitive work.

Details of the tool slides are shown by Figs. 66–69, which may be compared with the photographs. The cross slide *A* receives the saddle *B*, fitting against its face and having veed edges, the upper one with a gib and clearance. The lower portion is a narrow guide with a long take-up strip, adjusted from one end with a screw (Fig. 68). The saddle receives the swivel base *C*, pivoting on a central stud, and being clamped at any angle with bolts having tee-heads, moving round an annular tee-groove in the saddle. In the swivel base the turret slide *D* is fed vertically or at any angle with a pinion and rack. On it the turret *E* is rotated and locked at either of five positions.

The feeds are transmitted from the gears at the right-hand end of the cross slide through the rods *F* and *G*. The first is a screw that effects the cross traverse of the saddle, the second is splined to carry a worm whence the vertical and angular feeds are derived. In the duplex machines (Fig. 65) these fittings are duplicated for each head. For the screw traverse, *F*, a double bush *H*, with lock nuts to take up wear forms the nut (Fig. 68). For the down feed, the splined worm *J* partaking of the movement of the saddle, meshes with a worm wheel *K*, having a pinion *L* on its shaft that engages with a rack *M* on the vertical slide. The fitting of the slide with gibs is shown in the plan view, Fig. 67, and the clamping of the turret, and Fig. 68 shows how the turret is locked with a spring plunger, the levers for clamping and locking being seen in the photographs.

Above each of the turret slides in the photographs a circular box is seen, from which a wire rope descends. This is a counterbalance to the weight of the heads, formerly done in a clumsy fashion with a suspended weight. The new device is shown in section by Fig. 70. A coiled spring is enclosed and anchored in a box, its tension being adjusted for different loads with worm gear. A wire rope, fastened to a pulley and passing over it, descends through an opening in the box to the top of the vertical slide. The mechanism is carried on a light casting bolted to the top of the swivel base.

GEARING

By Henry E. Merritt, D.Sc. (Eng.)

LESSON VII

WORM GEARING

Worm gearing comprises two general types, namely, the "parallel" and the "globoidal" types respectively. It is proposed to deal in detail only with the former.

Tooth Action. A parallel worm has threads of truly helical form, i.e. the worm is actually a screw. It engages with teeth on the rim of a worm-wheel, so that on rotating the worm the threads by their screw action cause the worm-wheel to rotate.

When considered in detail it becomes apparent that the tooth action corresponds exactly to that of a rack and pinion, represented by the worm and the wheel respectively. For consider any section of the worm parallel with its axis, as shown in Fig. 36. Owing to the helical form of the worm, every adjacent thread has a similar section, spaced at equal distances apart, and therefore as the worm rotates the sections of the various threads simply advance axially without changing their form. There will thus be an imaginary straight line on the worm and a circle on the worm-wheel, corresponding to the pitch circles of a rack and pinion, and which make rolling contact at a point P. Measured round the pitch circle of the worm-wheel, the circular pitch of the teeth is equal to the axial pitch of the worm threads.

Pitch Diameters. From the above it follows that wherever the section of the worm be taken, the pitch circle of the wheel will have the same diameter and all the pitch points P collectively form a line PP (Fig. 37), which is the true "pitch line" of the worm gear. Thus, the wheel has a pitch cylinder, but the worm has not, the pitch surface of the latter being a plane containing the line PP. The pitch diameter of the worm is, therefore, the diameter of a circle which touches the pitch plane. A worm has, obviously, no pitch diameter or pitch plane when apart from the wheel, whilst when it engages with a wheel having a given number of teeth the position of the pitch line is fixed only relative to the wheel, and may vary relative to the worm according to the centre distance. For convenience in detail design it is customary to assume a nominal pitch diameter for the worm, which it is important to remember is not necessarily equal to the actual pitch diameter when in mesh with the wheel, although this is frequently the case.

Notation for Design. The D.B.S. standard notation for worm gearing is closely analogous to that for spiral gears, and is as follows—

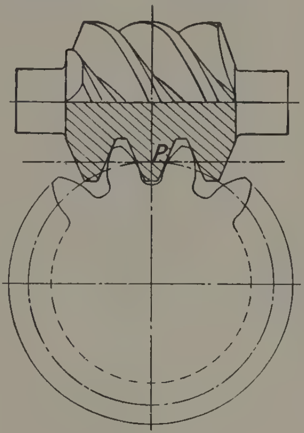


Fig. 36

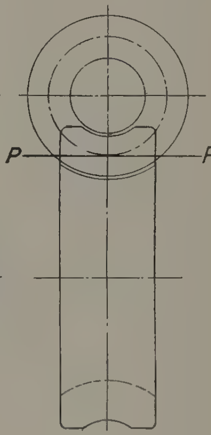


Fig. 37

C = centre distance.
d = pitch diameter (nominal) of worm.
D = pitch diameter (nominal) of wheel.
t = number of threads in worm.
T = number of teeth in wheel.
λ = lead angle of worm.
σ = spiral angle of worm.
ψ_a = axial pressure angle of worm threads.
L = lead of worm thread.
p_a = axial pitch of worm threads.
a = addendum of worm.
b = dedendum of worm.
o = overall diameter of worm.
i = root diameter of worm.
A = throat addendum of wheel.
A^1 = rim addendum of wheel.
O = throat diameter of wheel.
O^1 = overall diameter of wheel.
f_w = face-width of worm.
fW = face-width of wheel.

This notation is illustrated in Fig. 38.

General Formulae. Considering first the worm, the lead, pitch diameter and lead angle are connected by the relation

$$\tan \lambda = \frac{L}{\pi d}$$

and the lead and pitch by

$$L = t \, . \, p_a$$

The nominal pitch diameters and centre distance are connected by

$$C = \frac{d + D}{2}$$

but the actual pitch diameter D^1 of the wheel is

$$D^1 = \frac{T . \, p_a}{\pi}$$

and of the worm

$$d^1 = 2C - D^1$$

Detail Design. According to the conventional method of designing worm gearing, the axial section of the worm is regarded as an involute rack and is made straight-sided with an axial pressure angle of 14½ or 15°. The nominal pitch diameter of the worm is selected in a purely arbitrary fashion, and the thread proportions are made equal to those of standard spur gears. The formulae for detail design thus become

$$\begin{aligned} a &= A = \cdot 3183 pa. \\ b &= B = \cdot 3683 pa. \\ A^1 &= 1 \cdot 5a \\ o &= d + 2a. \\ O &= D + 2a. \\ O^1 &= D + 3a. \\ i &= d - 2b. \end{aligned}$$

If the lead angle is greater than 20° it is customary to base the tooth proportions on the normal pitch—

$$p_n = p_a \cos \lambda.$$

Disadvantages of Conventional Design. The conventional method of design is a very crude adaptation of spur gear proportions to the totally different requirements of worm gearing and results, in many cases, in very inefficient design. The disadvantages attendant on the conventional method are seldom realized, for this requires a more intimate knowledge of worm gear action than most designers possess.

These defects arise directly out of the standardization of the worm thread. Before discussing them, however, it may be remarked that standardization of this element in design, apart from the inferior tooth performance to which it leads, does not bring in its train any of the advantages which it is the object of standardization to secure. In the production of a worm-wheel, the hob required is similar to the worm in general dimensions, and therefore standardization of the thread form alone does not reduce the stock of hobs necessary and does not result in any saving. The " standard " thread section, however, also leads to worm gears having a relatively small load carrying capacity, and at the same time frequently gives rise to the defects of " undercutting " and " assembly interference."

Undercutting consists in the removal by the hob of material at the roots of the worm-wheel teeth to such an extent that the strength and load carrying capacity are materially reduced. It is similar to the undercutting which occurs with 14½° pressure angle spur gears, and is due to the same cause. With a straight-sided worm thread section, the wheel teeth on the axial section are involute, and correspond to a spur pinion. Undercutting then commences when the wheel has less than thirty teeth, and the proportion of undercut designs is undesirably large. Undercutting may be remedied in two ways, firstly, by applying correction, which consists in increasing the centre distance, other things being equal, and secondly, by increasing the axial pressure angle of the threads. In many cases this increase in pressure angle is an advantage, but no hard and fast rule can be laid down for the choice of pressure angle, owing

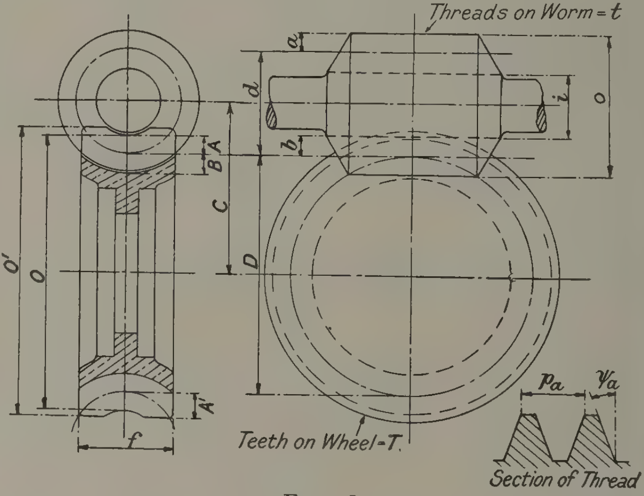


FIG. 38

to the many other factors which enter into the question.

Assembly Interference. A more obscure phenomenon, to which the term "assembly interference" may be given, is often met with when worms having a large lead angle are injudiciously designed. It is found that when the wheel has been hobbed with a hob of exactly the same form as the worm, the latter will not assemble with the wheel laterally, although it may be screwed into position endwise. When assembly fouling occurs the effective load carrying surface of the wheel teeth is materially reduced. The cause is found in a combination of a small pressure angle and large lead angle. On the central plane the thread section is symmetrical, but on other sections the profile changes, and a point is reached where the slope of the profile becomes negative, forming a kind of a dovetail which prevents assembly.

The D.B.S. Standard System of Worm Gear Design. In spite of the short space available for discussion of worm gearing, the present article would not be complete without reference to the most recent development in worm gear design, namely, the D.B.S. standard system of worm gear design. This system embodies the D.B.S. patent form of worm gear which, by reason of its geometrical form, eliminates undercutting and assembly interferences, and leads to an increase in load-carrying capacity as compared with the straight-sided type.

The underlying principle of the system consists in the adoption, not of a standard form of worm thread profile, but of a range of standard worm forms. In this way the arbitrary selection of a worm pitch diameter is eliminated, and the range of tools necessary is in consequence reduced.

The pitches used are based on the module system, which particularly lends itself to worm gear design. The standard worm forms are based on a standard range of pitch diameters for each pitch; these are arrived at by making the (nominal) pitch diameter a multiple of the module, this multiple being termed q. That is to say, the pitch diameter is equal to that of a spur gear having q teeth, and the centre distance is equal to that of a pair of spur gears having T and q teeth respectively. This leads to the fundamental relation

$$2C = M(T + q)$$

where M is the module. From this equation the values of C, M, T and q can be adjusted in order to enable a standard worm to be utilized.

The Efficiency of Worm Gearing. Well-designed worm gearing has a very high efficiency, values of 95 per cent and over being common. The approximate expression for calculating efficiency is

$$\eta = \frac{\tan \lambda}{\tan (\lambda + \phi)},$$

where ϕ is the angle of friction and η the efficiency. The coefficient of friction to be used in this expression ranges from ·015 to ·05 under ordinary circumstances.

The coefficient of friction depends both upon the lubricant and the materials employed. The highest efficiency is obtained using castor oil as a lubricant, with a worm-wheel of special bronze and a worm of steel, case-hardened, ground and polished.

BALL AND ROLLER BEARINGS

By W. E. Baker, B.Eng.

LESSON V

ROLLER JOURNAL BEARINGS

Early forms of roller journal bearings were designed for long thin rollers. The experience with them, apart from their power saving qualities, was somewhat disappointing as the rollers had a tendency to become skewed and run off endways. Early in the present century the modern short roller bearing (see Fig. 22) was evolved. Apart from being made to the same degree of accuracy as the ball journal bearing, this modern short roller bearing embodies the following features—

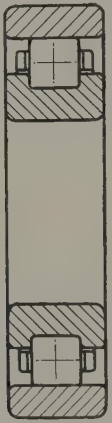


Fig. 22. Roller Journal Bearing
With short cylindrical rollers

1. The length of the roller does not exceed its diameter.

2. An accurately machined channel is provided in the inner race, which serves accurately to guide the roller and prevent any tendency to skew.

3. There is no channel in the outer race which is, therefore, free to slide over the rollers and take up any endways position.

The rollers and races are made from special specifications of steel, suitably hardened and ground. A yellow metal cage is interposed between the rollers in the same way as is done in the ball journal bearing.

Mounting. The revolving race of the roller journal bearing is made a tight fit onto (or into) the revolving member, for the same reason as was explained in Fig. 11, when dealing with the ball journal bearings.

With regard to the stationary race, the practice differs from that obtaining with ball journal bearings. In the latter, as explained by means of Fig. 12, it was necessary to make the stationary race a push-fit in the stationary

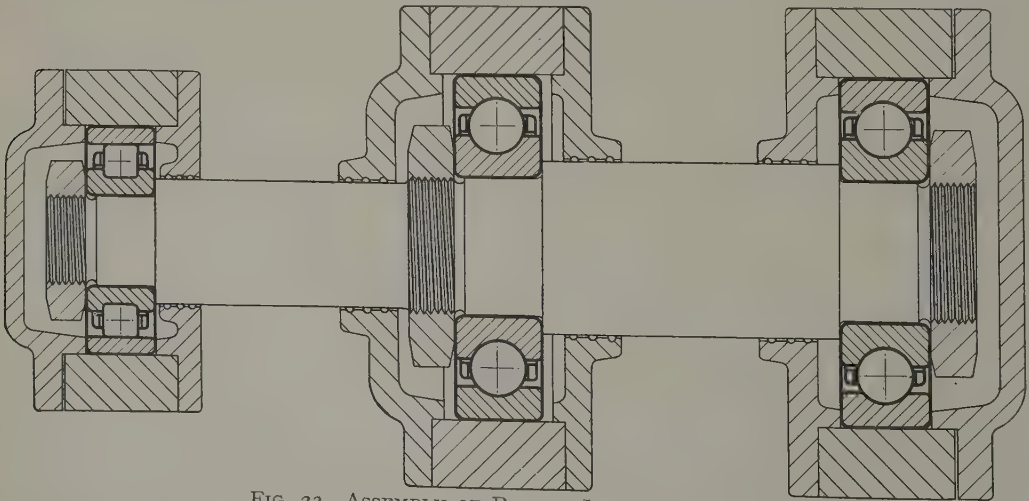


Fig. 23. Assembly of Roller Journal Bearing
Compared with unlocked and locked ball journals

member. With the roller bearing, on the other hand, there is no groove in the outer race and there is, therefore, no danger of nipping the rollers endways. The stationary race of the roller bearing may, therefore, also be made a tight fit to the stationary member. In any case, it will be necessary to position both races of the roller bearing endways, and it is not necessary that the rollers should track accurately in the centre of the outer race.

These points will be made clear by reference to Fig. 23. This figure represents a revolving shaft supported on two ball journal bearings and one roller journal bearing. It is assumed that no end-thrust is present. The inner races of all three bearings are made a tight fit on the shaft and are clamped up between shoulders and nuts. It will be noticed that the face of the nut against the roller journal is distinctly chamfered back, so that the nut is not in contact with the race at a greater diameter than the roller track. This prevents the channel in the inner race being nipped up on to the rollers. The danger of this happening is not so severe in the case of the ball journals, as the channel is stronger against lateral pressure.

The outer races of the ball journal bearings are made push-fits in the housings, and the right-hand one is shown gripped endways in order to locate the whole shaft in position, no end-thrust being present. The outer race of the centre bearing (which is a ball journal) is left free endways so that expansion can take place without danger of nipping the balls.

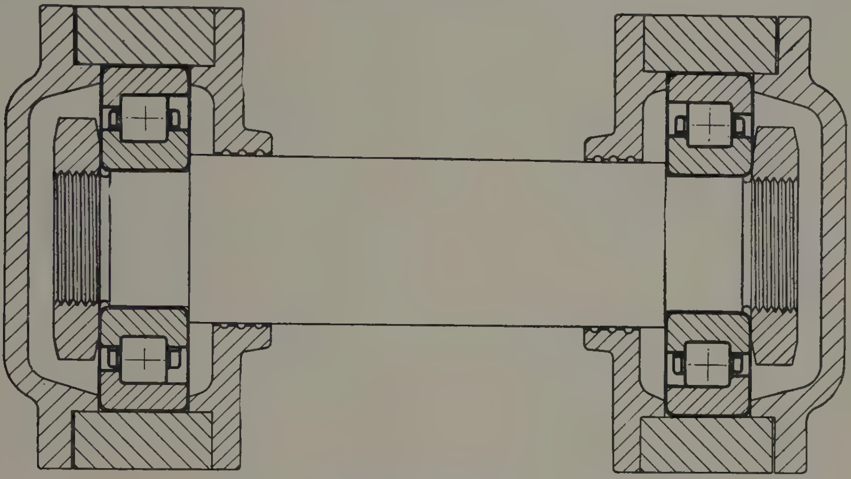


FIG. 24. ROLLER JOURNAL BEARINGS
With one lip in outer race

The outer race of the roller journal may be made a tight fit, and must be clamped endways as indicated. Any expansion reaching this bearing merely causes the rollers to slide through the outer race.

It will be noticed that end clearance is shown between the housings and the flanges of the left-hand covers in both the outside housings, indicating that the outer races in those housings are actually gripped.

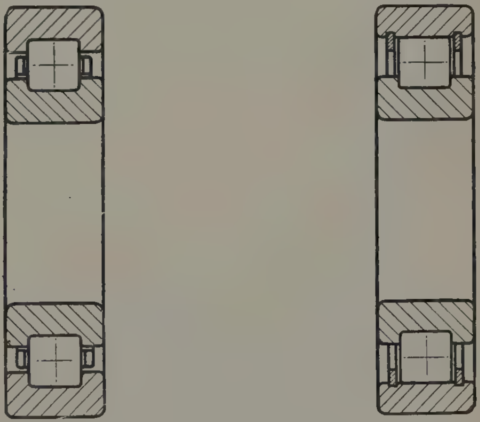


FIG. 25. ROLLER JOURNAL BEARINGS
With
Two lips in outer race — Retaining rings in outer race

Apart from this difference, roller journal bearings are treated similarly to ball journal bearings in the methods of fitting and lubrication.

THE ADVANTAGES OF ROLLER JOURNALS OVER BALL JOURNALS

Roller journal bearings differ from ball journal bearings in making line contact with the races instead of point contact. This enables them to deal with a greater proportion of shock than is possible in the case of ball journals.

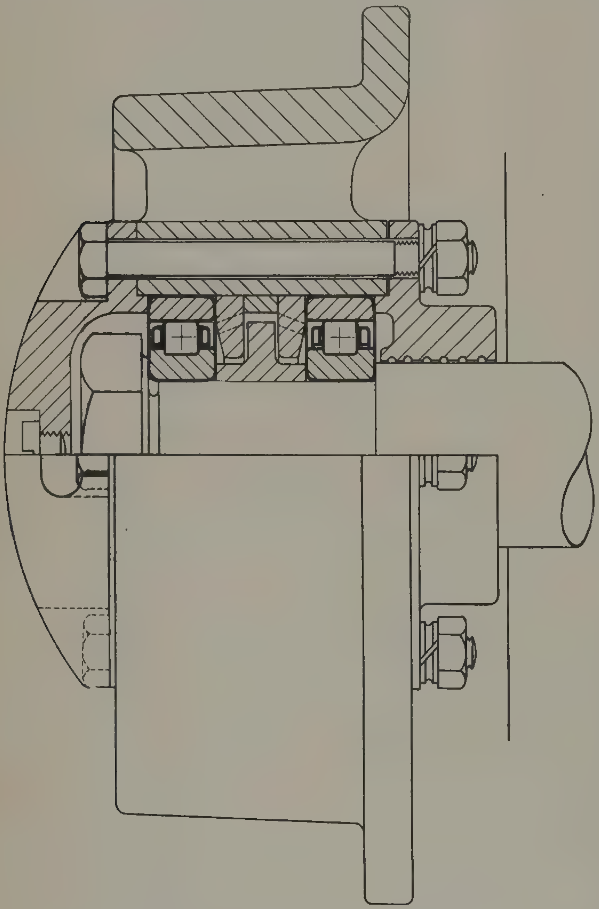


FIG. 26. ROLLER BEARING WHEEL
Showing plain surfaces for endways location

Moreover, size for size, they are capable of carrying heavier loads, the increased capacity being 50 to 200 per cent more, according to the actual duty involved.

Contrary to expectation, the coefficient of friction of the precision short roller journal bearing is not greater than that of the ball journal bearing, and in fact under heavy loads, is actually less. On the other hand, under very light loads the roller journal bearing is somewhat more sluggish than the ball journal bearing.

Apart from the above, the short roller journal bearing has an advantage in having no channel in the outer race. As already explained, this enables both races to be made a tight fit respectively to the revolving and to the stationary parts. If an out-of-balance load is met where ball journals are used, this will cause the stationary race (which has been made a push-fit) to creep on its seating. It is not possible to make the stationary race of the ball journal bearing a tight fit on its seating and, therefore, the effect must be allowed to take place. With the roller journal bearing, on the other hand, the stationary race may be made a tight fit in its seating, and this will cure creep of the stationary race, due to out of balance, in just the same way that the tight fit of the revolving race cured creep of the revolving race. This does not mean that roller journals should be subjected to undue out-of-balance loads, but it is distinctly helpful in the case of vertical spindle mountings where the downwards weight does not assist the journal bearings to maintain their seats against out-of-balance load.

OTHER CONSTRUCTIONS

The short roller bearing is sometimes made with one lip in the outer race, as shown in Fig. 24. The object of a design of this type is to provide an easy method of locating the shaft endways, where no end-thrust is developed. Great care is necessary when fitting up to ensure that the rollers are not nipped endways.

Examples are also found of short roller bearings having channels in both races, as shown to the left of Fig. 25. These bearings are fitted in exactly the same way as ordinary ball journal bearings. The same applies to the bearing shown at the right of Fig. 25, in which retaining wires are fitted in the outer race of the roller journal bearing. In this case the bearing is not usually provided with a cage.

OTHER METHODS OF LOCATION

The most positive method of holding a revolving member endways is by means of the double-thrust bearing, shown in Fig. 20. However, where no thrust is developed, it is not desirable, nor necessary, to go to this expense.

In dealing with ball journal bearings, Fig. 13 has shown how location may be effected on one of the ball journal bearings. However, the introduction of roller journal bearings brings this question of location again before us, inasmuch as Fig. 23 has already illustrated the inability of the ordinary roller journal bearing to cope with location duties.

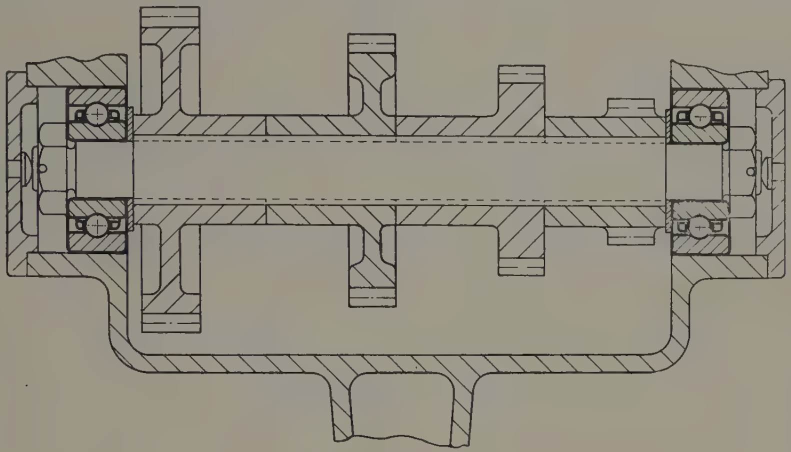


FIG. 27. BALL BEARINGS ON LAYSHAFT OF GEAR-BOX
Showing buttons for location

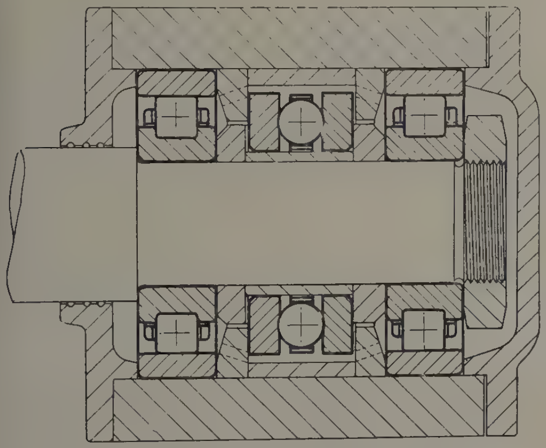


FIG. 28. ROLLER JOURNALS
With Single Ball Thrust arranged to take thrust in both directions

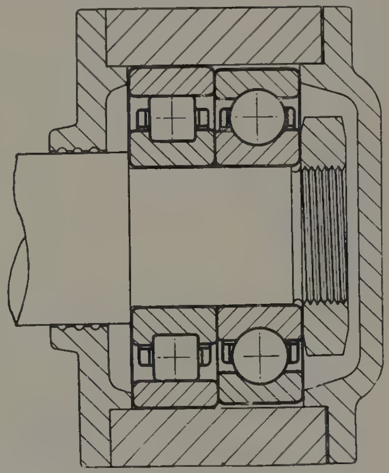


FIG. 29. ROLLER JOURNAL
With ball location bearing

In the same way, Fig. 23 has indicated one solution of the problem, namely, to use one ball journal bearing in each mounting and to effect location at that point. This is a useful method where possible, and is adopted on medium size electric motors, the roller journal being used at the driving end and the ball journal at the opposite end. Another method is to arrange plain surfaces to deal with the location duty. Fig. 26 shows an arrangement of such surfaces within the boss of a wheel mounted on roller journal bearings. The axle is stationary and the whole wheel revolves about it.

Fig. 27 shows another method in which hardened steel buttons are used at the end of the shaft to effect location. This is commonly met with on the lay shaft in motor-car gear-boxes.

Another possibility is to arrange a single-thrust bearing to be capable of taking thrust in both directions, as shown in Fig. 28. As shown, the revolving shaft is endeavouring to move from right to left, the right-hand race of the thrust race revolving and the left-hand race being stationary. If the shaft endeavours to move from left to right the clearances are taken up, and the left-hand race of the thrust bearing becomes the revolving one and the right becomes stationary. The method is clearly of limited application, and could not be used if the thrust load was of a rapidly alternating type.

A further method, where the thrust load is not too heavy, is to adopt a special " location " bearing, as shown in Fig. 29. The " location " bearing is similar to the ball journal bearing, but is of the deep groove variety without filling slots at the side. Moreover, it is made smaller than nominal dimension on its outside diameter. By this means it cannot be subjected to journal loads because it is clear in the housing. Under these conditions all the balls are free to share the thrust load, for which purpose it is designed. It is an easy bearing to mount, requiring the same dimension both of shaft and housing as the roller journal alongside, and it is, moreover, a one-piece unit in itself. These " location " bearings are used for this duty on large electrical machines, and also on many wood-working spindles, including vertical moulding spindles. " Location " bearings are admirably suited for thrust loads within their capacity, especially at high speeds. Amongst other heavy duties, they have stood up successfully in practice to the end-thrust found in tramcar axle boxes.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. Workman, B.Sc., A.M.I.E.E., etc.

LESSON VI

THE DESIGNING DEPARTMENT

The Drawing Office. Before any article can be manufactured, it is necessary to give detailed instructions to the different workpeople involved. The most convenient way—and often the only way—of giving these instructions is by means of a drawing of the article to be made. Of course, the drawing is not the *only* instruction given, as specification of the material to be used and so on is also required. But usually the different drawings form the most important part of the instructions, and also take the greatest time to prepare. The responsibility for the preparation of these drawings, and for the form which they take, rests with the drawing office. The responsibility is a large one, as mistakes or low efficiency in the work of this department cannot usually be rectified at a later period in the manufacture. On the other hand, a high efficiency in this department, involving the practical development of high-class new ideas, and a constant application of brain power in the elimination of waste and the production of economy, will have the greatest effect in the success in the business, and will help to produce similar advantages in the actual manufacturing.

Conditions Necessary. In order that the drawing office may produce its best work it is obviously necessary to provide the department with the best conditions. The chief of these conditions are as follows—

1. *Quietness.* The quality of the work of a designer largely depends on the amount of concentration that he is able to give to his work. The more continuous that his attention is to any particular work, the more likely is it that the work will be good. For this reason the drawing office should be situated in as quiet a portion of the works as is available, and visitors to it of all sorts should be discouraged as much as possible. At the same time the need for quietness must not be considered alone without reference to the almost equal necessity of the drawing office being moderately handy to the

shops and general offices, otherwise designers will either lose touch with the practical application of their work, or will be constrained to spend an undue amount of time in walking about the works.

2. *Light.* The second main consideration in the choice of the location of the drawing office is the question of light. As regards natural light, large windows should be provided, preferably with a northern aspect. The drawing boards should be arranged so that the light falls on the drawing over the left shoulder of the draughtsman. As regards artificial lighting, individual electric lights are probably the best, as they permit of individual adjustment for each draughtsman.

Filing. The question of the provision of good filing accommodation for drawings and tracings must be carefully considered. The quantity will often be very large, and the individual sheets of a very inconvenient size, from a filing point of view. The files should be in a separate room close to the drawing office, and in the charge of a separate staff. This staff alone should have access to the drawings, and should be responsible for all issues and receipts. By means of small cards, a record should be kept of the issue of all drawings so that they may be found when wanted. Drawings should only be issued on the authority of certain recognized officials.

Staff. The organization of the personnel of the design office will vary somewhat in different works. Usually, the following staff will be included in the organization of this department.

Chief Engineer.
Designers.
Tracers.
Chief Draughtsman.
Draughtsmen.
Estimators.

On the other hand, it may be found advisable to organize the chief engineer and designers as a separate department from the drawing office, although their work will always be closely related.

The Chief Draughtsman. This person is the head of the drawing office, and must have sufficient technical knowledge to be able to control and criticize the work of his staff. The personal link between the two will often be necessarily much greater than in the case of the head and staff in other departments, as so much depends on his personal touch and inspiration.

Designers and Draughtsmen. The designer provides the inventive qualities of the department, and is responsible for materializing new ideas and developments into practical form. The draughtsman, however, does not require as much imagination, as his work is to express in a drawing the ideas and calculations of the designer. At the same time, he will often be responsible for the detailed design of the smaller parts, and should thus have a good knowledge of the strength of materials and of existing standard design of parts. When the drawings have been completed and checked, the taking of the tracings for the obtaining of "blueprints" may be done by unskilled staff, whose chief qualifications are neatness and accuracy in copying.

The Checking of Drawings. It is essential that no drawings should be allowed to be issued until a systematic check has gone as far as possible to eliminate every error, as a slight error may be the cause of a very large expense, such as the scrapping of a casting costing hundreds of pounds. The check should be made with the following points in view—

1. All dimensions must agree with each other.
2. All dimensions necessary for manufacture must be quoted.
3. Dimensions must be put on the drawing according to a definite scheme, duplication of figures being avoided so as to eliminate the possibility of an alteration being made to one figure and not to another.
4. Standard parts must be used wherever possible instead of new designs.
5. The "finish" required should be clearly stated where necessary.
6. Where there are alternative designs possible, the one to be chosen must be the one which is easier and more economical from a manufacturing point of view, and which will cost the least.

Standardization. The question of standardization must always be carefully considered by the members of the drawing office, as it is largely upon their actions that the completeness of its adoption depends. Items, such as bolts, nuts, screw-threads, and so on, have already been standardized irrespective of any particular factory. Their adoption being general, they should always be specified where possible, both to limit the varieties manufactured and to enable customers to replace easily. In addition to this class of item there will also be a large number of tools and parts which can be standardized for use in the particular factory concerned. It is of the utmost importance that when a standard or line of standards has been decided upon, a full record should be made in an available form. Unless the record is an efficient one it will soon be found that standard parts are being duplicated. Standardization must be particularly watched in any factory making a line of goods in different sizes. The tendency is to use a different size for each part when the size of the complete article varies, the result being a great increase in the quantity of stocks carried, and a greater cost due to the smaller amounts of each type and size manufactured.

Blue-prints. In order that a drawing may be duplicated and also put into a better form for preservation, "blue-prints" are usually made photographically from tracings of the drawings. If an alteration is made to the original drawing it is obviously very necessary to ensure that all copies are similarly altered, and this emphasizes the necessity of keeping an efficient record of the issue of all drawings and copies.

Arrangement of Drawings. The complete design of even a moderately complicated piece of machinery is so complex that simplification of the drawings has to be obtained by means of subdivision. The three classes of drawing in common use are—

1. *The General Arrangement.* This is more of an erection drawing, and only shows the relative positions of the different parts.

2. *The Detail Assembly Drawing.* This type of drawing gives sufficient information for the different detail parts to be assembled into sections.

3. *The Detail Drawing.* This drawing is made for each part, or group of parts, is fully dimensioned, and gives all particulars necessary for manufacture.

Advantages of Subdivision. Although the number of drawings is greatly increased by subdivision, there are several advantages which more than compensate for this—

1. *Less drawing,* as parts appearing in different machines need only be drawn once.
2. *Fewer manufacturing mistakes,* as working drawings are enormously simplified.
3. *Handier size* as detail drawings will be much smaller.
4. *Standardization* is more easily attained.
5. *Alterations* are more easily and cheaply made.

Headings of Drawings. As certain particulars are always required on a drawing, a form for this purpose is often printed on the tracing paper so that it may be reproduced without extra tracing. The particulars usually required are as follows—

1. Name of firm.
2. Name of customer (if not for stock).
3. Name of part.
4. Drawing number.
5. Scale of drawing.
6. Date of drawing.
7. Dates of alterations.
8. Initials of those responsible for the design, drawings, tracing, checking, and altering of the drawing.

PATENTS FOR INVENTION

By W. E. Dommett, Wh.Ex., M.I.Mar.E., A.F.R.Ae.S.

LESSON V

THE COMPLETE SPECIFICATION FORM 3

The complete specification must be filed in duplicate, and the first page should be on Patents Form No. 3. One form should be marked "Original" at the top right-hand corner, and the other "True Copy." The "Original" form needs a stamp of £3. The filling in of this form is quite simple, in the space (*a*) is inserted the title of the invention which must be identical with that on Form 1; in the space (*b*) should be inserted the full name, address, and nationality of the applicant or applicants, and again this must be identical with the corresponding entry in Form 1.

In cases where a provisional specification has already been filed the official number and date thereof must be put in the space clearly provided for it. In this case the complete specification, both original and true copies, will be filed at the Patent Office without any other documents, but if no previous provisional specification has been filed then it is necessary to send at the same time an application form, No. 1, properly filled in, as indicated in Lesson II of this series.

Section 2 of the Patents Acts, Subsec. 2, requires that: "A complete specification must particularly describe and ascertain the nature of the invention and the manner in which the same is to be performed," and accordingly there is a declaration printed on this form, below the name and address, to the effect: "Do hereby declare the nature of this invention and in what manner the same is to be performed, to be particularly described and ascertained in and by the following statement."

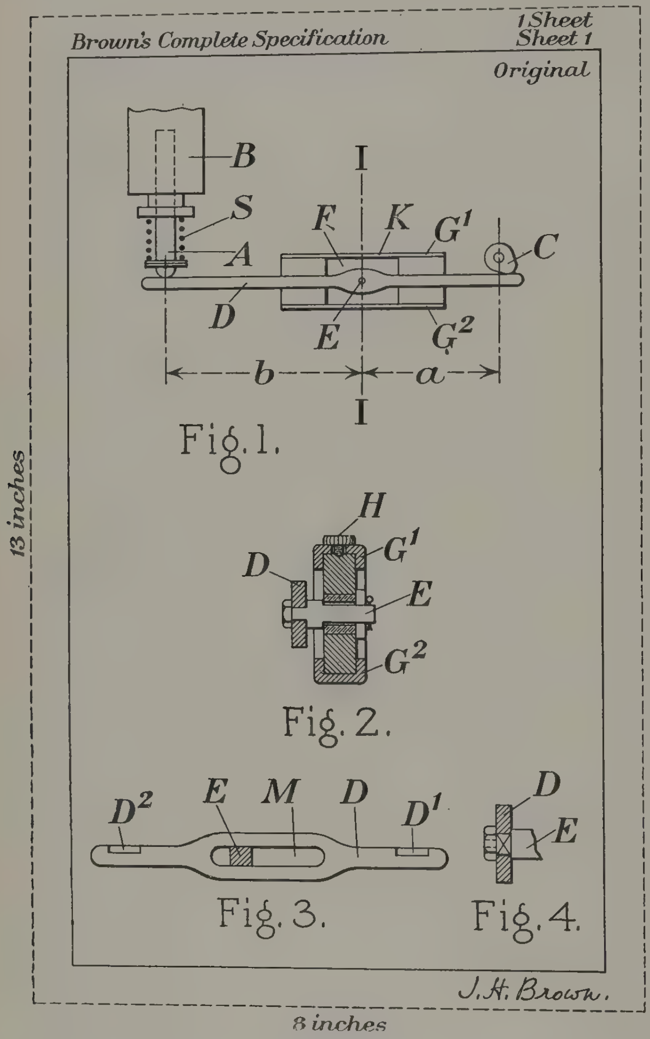
As in the case of Patents Form 2 there is a space at (*c*) in which this statement could be started, but it is desirable to leave this blank and to make the actual start of the statement on a new foolscap page, which should be numbered page 2. To save repetition it is proposed to utilize the example relating to variable stroke fuel pumps in the present section, and accordingly the opening clause, second, third, fourth, fifth, and sixth clauses, can be repeated to form the opening part of the complete specification. In so doing we have complied with the first essential in that we have set out the nature of the invention.

In order to comply with the rest of the official requirement it is necessary now to provide a particular description, so that the next clause should be somewhat on the following lines—

"In order that the details of construction and action of my invention may be the more clearly understood, reference is made to the accompanying drawings, in which Fig. 1 represents a side view of the actuating gear, Fig. 2 a section on the line *II* of Fig. 1, Fig. 3 represents a modification of Fig. 1, and Fig. 4 a section corresponding to Fig. 2.

"In carrying my invention into practical effect, I arrange to actuate the plunger *A* of a fuel pump *B* by a cam *C* through the intermediary pivoted lever *D*. The lever *D* has rigidly secured therein a fulcrum or pivot pin *E*, which is adapted to rest in a bushed bearing in a block *F*. The block *F* is positioned between upper and lower guide members G^1G^2, and is fixed in any required position therein by a set-screw *H* passing through a slot *K* in the upper guide G^1, as shown in the drawing. The distances of the fulcrum *E* from the centre line of the cam *C* and line of stroke of the plunger *A* are *a* and *b* respectively, so that the stroke of the plunger *A* is $\frac{a}{b}$ times the lift of the cam. It will be obvious then that if the block *F* and, therefore, the fulcrum is shifted so as to vary the values of *b* and *a*, then the stroke of the plunger *A* will also be varied. As can be seen, the movement of the block *F* is rendered easy of accomplishment, since it only needs slacking back the set-screw *H*.

"Where definite variations and not infinitely variable variations are required, the slot *K* could be dispensed with and a number of holes used instead, which would obviate any shifting of the block should the set pin become slack, but which necessitates the complete withdrawal of the pin when each readjustment is made. It should be noted that with this construction, different portions of the ends of the lever *D* are in contact with the plunger *A* and the cam *C* with different fulcrum settings, and that wear of the lever ends is reduced accordingly. However, it is obvious that a longer lever is necessary than with a fixed fulcrum arrangement. Should it be desirable, say, for reasons of space, to keep the lever as short as possible, I modify the arrangement as indicated in Figs. 3 and 4. In this form of construction the lever *D* is slotted at *M*, and the wrist pin *E* has a squared end entering this slot with a screw and nut for securing the two together. In this form this screw and nut connection is slacked back when the block *F* is to be adjusted, so that movement of the block results in the pin *E* sliding in the lever. When in the desired position the block in its guide and the pin in the slot are both resecured. Since the lever itself is not slidden endwise it need not be so long as in the previously described construction. I may provide the lever with renewable surfaces D^1, D^2, at the positions of contact with the cam and the plunger respectively.



"It will be seen that no lost motion between the lever and the cam and plunger can occur as long as the return spring *S* of the plunger is functioning.

"It is to be understood that I do not wish to limit myself to the details hereinbefore described, but may vary them as required without departing from the spirit and scope of the invention."

The statement of the invention claimed with which a complete specification must end, see Rule 14, shall be clear and succinct as well as separate and distinct from the body of the specification, and a formal preamble to the claims is necessary so that we continue as follows : "Having now particularly described and ascertained the nature of my said invention, and in what manner the same is to be performed, I declare that what I claim is—

"1. A variable stroke fuel pump characterized by an adjustable fulcrummed lever between the plunger and driving cam or eccentric.

"2. A variable stroke pump as in Claim 1, wherein the fulcrum is supported in a block slidably mounted in a guide or guides.

"3. A variable stroke pump as in Claims 1 and 2, in which the fulcrum pin is non-adjustable on the lever.

"4. A variable stroke pump as in Claims 1, 2, and 3, in which the fulcrum pin is adjustable in a slot in the lever.

"5. A variable stroke pump as in any of the preceding claims, wherein the lever is provided with renewable contact pieces.

"6. A variable stroke pump substantially as herein described and illustrated in the accompanying drawings."

Then follows the date, and signature of the applicant or his agent.

Care should be taken to see that the claims do not include either more or less than it is desired to protect by the patent.

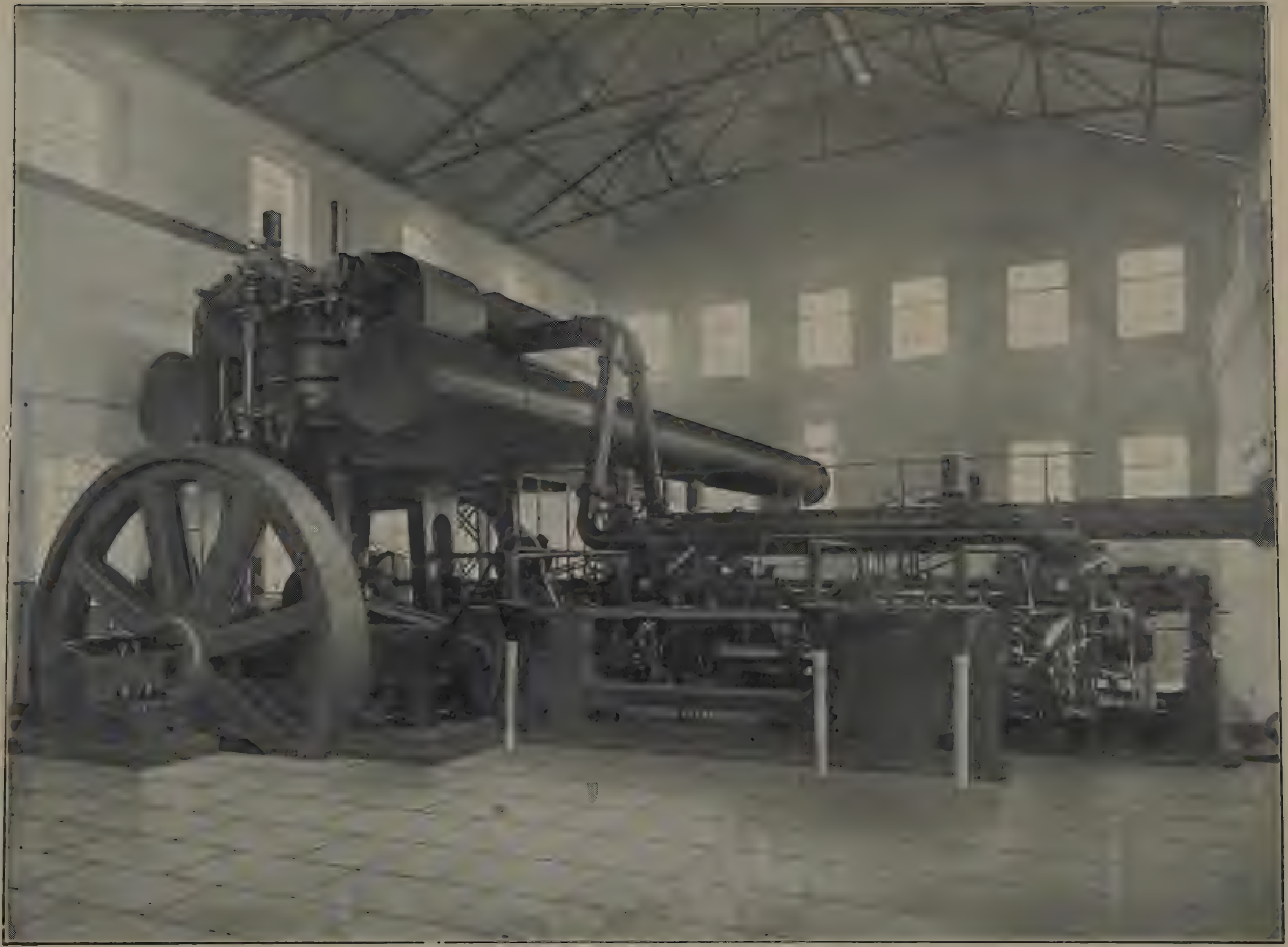


FIG. 32. 2,000 H.P. "PREMIER" BLOWING ENGINE

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON VII

REPRESENTATIVE ENGINES —(*contd.*)

Vertical Engines. While the horizontal type of gas engine is probably the more usual, it must not be overlooked that several makers manufacture equally good and efficient vertical engines.

Circumstances may decide which type is more suitable in any particular instance. Generally speaking, the horizontal type is more accessible and all working parts are near ground level. It does, of course, require more floor space though not as much head room as the vertical engine. At the same time, the latter can be of simple and compact construction, and be reliable and economical in operation. For high rotative speeds, such as those required for electric generation, etc., forced lubrication and an enclosed crankcase are necessary, and the vertical type of engine is then usually preferable. Single cylinders may, of course, be arranged side by side, but another scheme is to have two cylinders placed tandem over each crank, there being 2, 3, 4, or 6 of the latter. This construction makes for a cheaper and more compact engine, the reciprocating weights per cylinder are reduced and, while working on the four-stroke cycle, each crank receives an impulse every revolution.

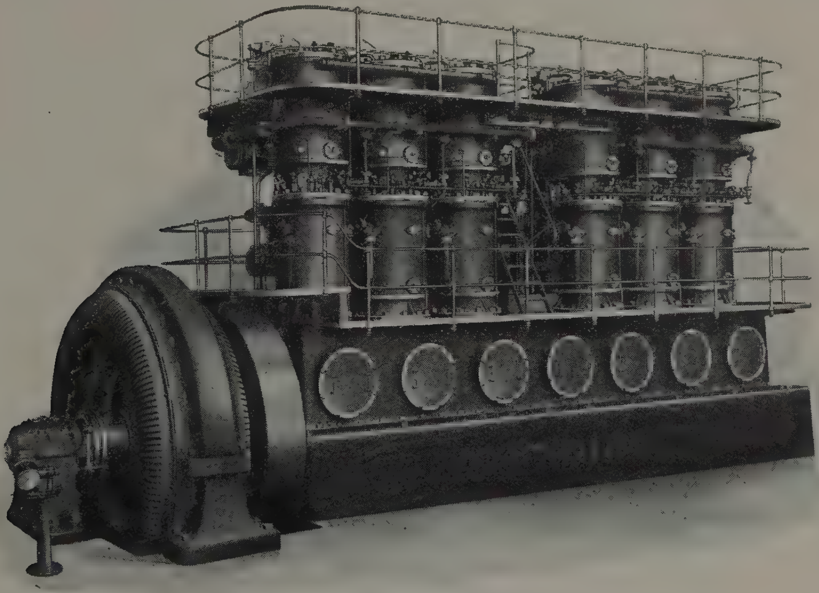


Fig. 29. 1,500 H.P. National Vertical Tandem Engine

Vertical Tandem Engine. By the kindness of the National Gas Engine Co., Ltd., particulars are here given of the engines of this type made by them, ranging in power from 300 to 2,000 h.p. The 1,500 h.p. engine is illustrated in Fig. 29, while sections of the 300 h.p. engine are shown in Fig. 30.

The bedplate is a single casting, except in the largest sizes when it is made in two parts bolted together with fitted bolts. The crankshaft bearings or "steps" are round backed (and so may be withdrawn without lifting out the crankshaft), and are secured by caps. The crankcase is securely bolted to the bedplate and is well ribbed internally. Large inspection doors provide easy access to the big ends, etc. The bearing next to the fly-wheel is made larger than the rest to allow for the increased working stress, while end-thrust is taken up by the collars provided on this bearing.

Each line of cylinders and pistons forms a complete unit, and all are interchangeable. The cylinder barrel, combustion chamber, and exhaust outlet branch are all water-jacketed. As will be seen from Fig. 31, the top cylinder is bored out slightly larger than the bottom one, and the joint between the latter and the "intermediate head" at this point is made by a bevelled copper ring. A number of "pole-screws" retain the intermediate head in position.

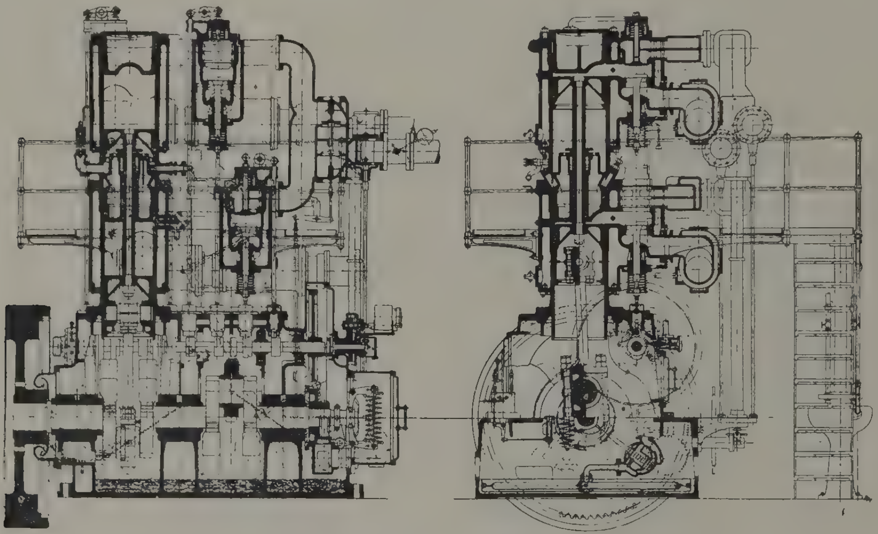


Fig. 30. 300 H.P. National Vertical Tandem Engine

The top head or cover is held down in the usual manner and both heads are water-cooled. The intermediate head carries a stuffing box in which a number of cast-iron spring rings make a joint with the piston-rod. The latter is of steel, and has a cast-iron sleeve forced over it, which prevents corrosion and makes for easier working in the intermediate head. It will be noticed that the bottom piston is of the usual trunk form, but the top one, having no side thrust to take, is made shorter.

The space between the top piston and the intermediate head is used as a "buffer" cylinder with alternate compression and re-expansion of the air contained therein. As one or other of the two pistons is compressing a charge on every upstroke, this means that all moving parts are "cushioned" on both up and down strokes.

The valves are operated through push rods, and those of the top cylinder are "staggered" relative to those of the bottom cylinder, thus facilitating removal.

Further details will be given in subsequent consideration of the subjects of governing, ignition, etc.

Large Engines. One of the factors which contributed to the development of very large gas engines was the discovery (due to Mr. B. H. Thwaite) that blast-furnace gases, the majority of which were usually wasted, could be successfully utilized in a gas engine cylinder. Advantage is now taken of this, in many iron and steel works, not only for the generation of electricity, etc., but also for the driving of the "blowing tubs" which supply the air blast to the furnaces. In Fig. 32 (Frontispiece) is shown a 2,000 h.p. engine (by the Premier Gas Engine Co., Ltd.) installed at the steel works of Messrs. Alfred Hickman, Ltd., Wolverhampton.

There are two sets of horizontal tandem cylinders side by side, the blowing tubs being placed vertically and driven direct from the cranks. The sizes are as follows—

	Engine Cylinders	*Blowing Cylinders*
Back	38 in. diameter × 48 in. stroke	79 in. diameter × 48 in. stroke
Front	39 in. diameter × 48 in. stroke	

The complete set is capable of delivering 40,000 cub. ft. of air per minute at a pressure of 7·5 lb. per sq. in., the engine running at 85 r.p.m.

A somewhat similar engine is constructed by Messrs. Galloways, Ltd., Manchester, but in this case the blowing cylinder is in line with the engine cylinders and the latter are double-acting (Fig. 33).

An installation of these engines, both for blowing purposes and also electric generation, at the works of the Barrow Hematite Steel Co., Ltd., is shown in Fig. 34.

The problem of unequal expansion of cylinder walls, present to some extent in all gas engines, becomes intensified in the case of very large engines. The cylinder must be sufficiently firm and strong to withstand the maximum possible explosion pressure without being so rigid that expansion, owing to temperature differences, is restricted or prevented. In the Galloway engine the cylinder, together with the valve pockets, is cast in two halves. These are forced hydraulically over a liner of hard cast iron having a central flange, which is nipped between the two cylinder halves when the latter are bolted together. The liner is thus held securely while being free to expand lengthways (Fig. 35).

The central portion of the water jacket is open in the casting and is closed by a light steel

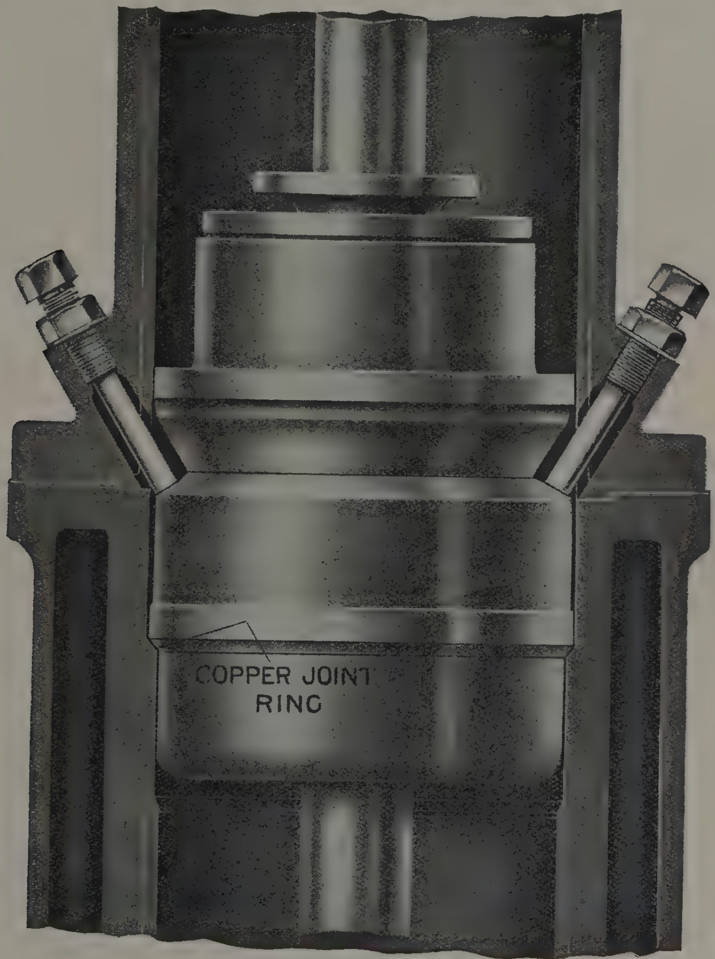


Fig. 31. Intermediate Head and Polescrews
(National Vertical Tandem Engine)

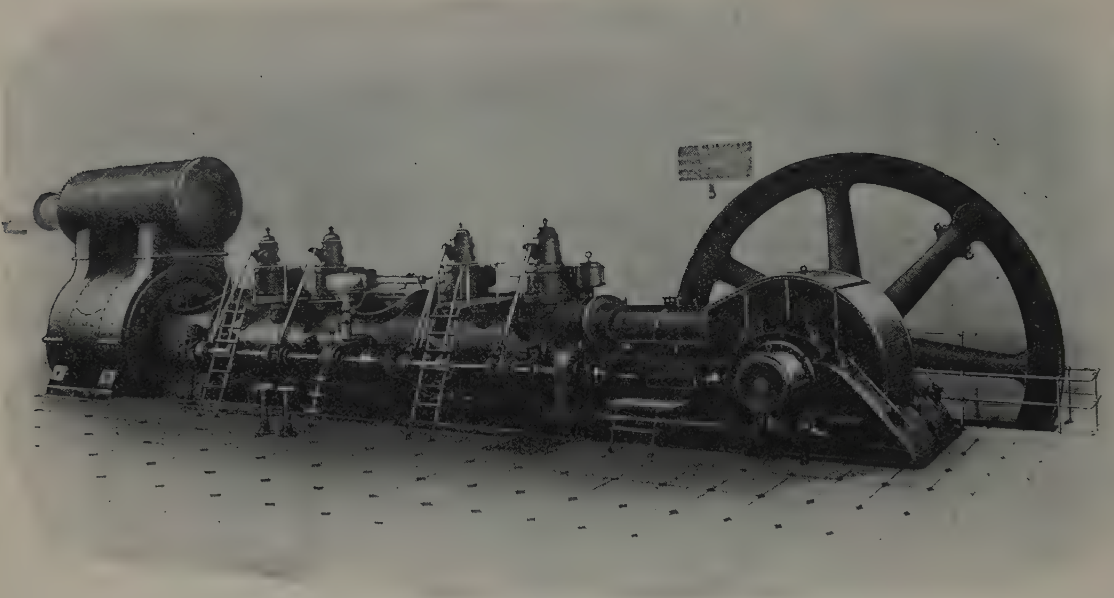


Fig. 33. 1650 H.P. Galloway Blowing Engine

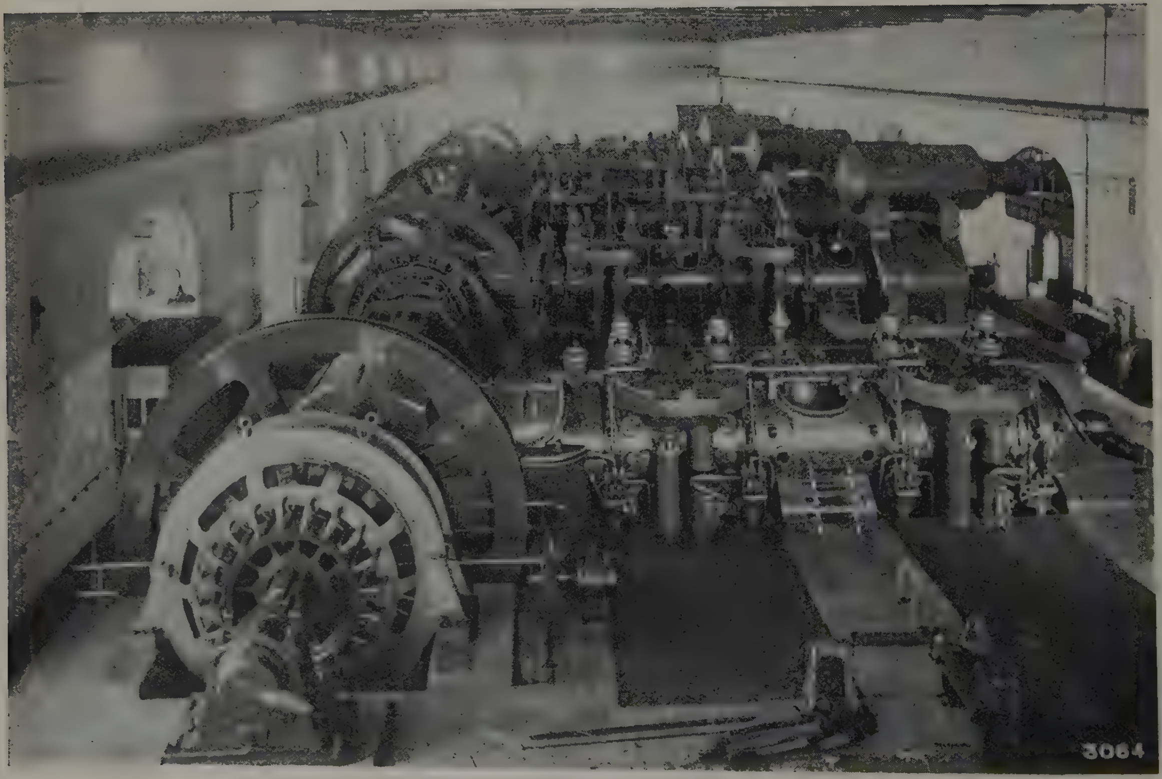


Fig. 34. Galloway Engines
(Barrow Hematite Steel Co., Ltd.)

jacket (consisting of two semicircular parts bolted together), the joint all round being made by rubber rings.

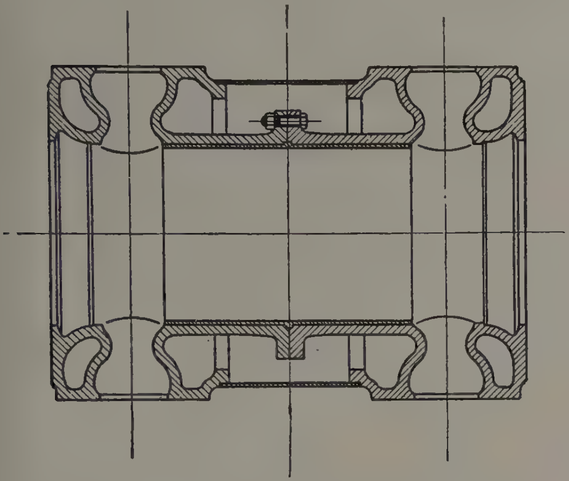
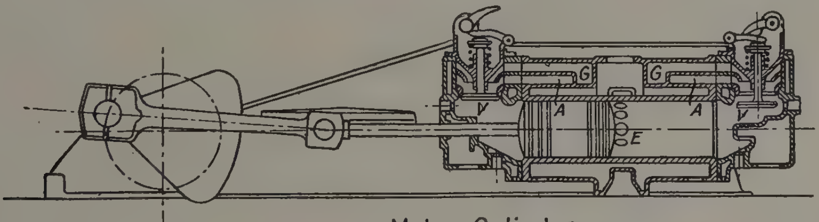
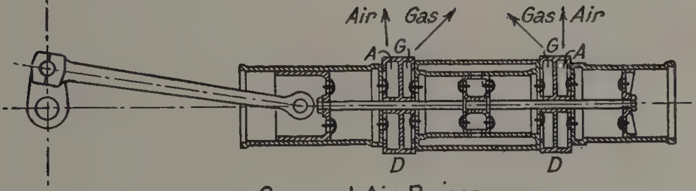


FIG. 35. BUILT-UP CYLINDER (GALLOWAY)

Two-stroke Engines. The Körting and Oechelhauser engines, which are modern developments of the Clerk engine, both originated in Germany. They were afterwards constructed, in improved form, in this country by Messrs. Mather & Platt, Ltd., and Messrs. Beardmore & Co., Ltd., respectively, but these firms no longer carry on this particular work.



Motor Cylinder



Gas and Air Pumps.

FIG. 36. SECTIONAL ELEVATIONS. KÖRTING ENGINE

A diagrammatic arrangement of one form of Körting engine is shown in Fig. 36. It is a double-acting engine and has a ring of exhaust ports *E* at the centre of the cylinder. The piston has a length equal to about 80 per cent of the stroke. Hence the exhaust ports are uncovered when the crank is about 40° from the dead centre, and are fully open in the latter position. Gas and air are admitted through mushroom valves *V*, which are mechanically operated through "rolling levers," and an eccentric on the crankshaft being connected together so that when one is open the other is shut.

The cylinder is bolted between girder frames which also carry the main bearings, crosshead guides, etc. Two single-acting air pumps with a double-acting gas pump between, are bolted on one side and driven by a small crank placed about 90° relative to the main crank. Delivery boxes *D* are placed between the pumps and all valves are automatic in action. Passages *G* and *A*, respectively, are provided in the engine frame for the gas and air which are kept entirely separate until they reach the inlet valve casing. The gas pump piston is hollow and has a number of slots in its periphery, while suction ports in the cylinder walls communicate with the gas supply. The latter may flow into the cylinder either direct or via the piston, according to the position of the latter, but gas will not be delivered to the motor cylinder until the suction port is covered by the gas pump piston, and hence the scavenging air will always have a "lead" over the gas. Governing is effected by causing some of the gas to be by-passed from one side of the pump to the other.

In the Oechelhauser engine, a diagrammatic arrangement of which is shown in Fig. 37, there are two pistons working in opposite directions in one cylinder. There are no valves (except a small one for starting purposes), both admission and exhaust being controlled by the uncovering of ports by the piston. The front piston is coupled by a crosshead and connecting rod to a crank. At 180° to this are two other cranks which are connected, through side rods and a special crosshead, to the rear piston.

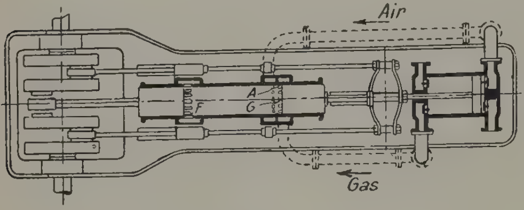


FIG. 37. DIAGRAMMATIC ARRANGEMENT OECHELHAUSER ENGINE

Assuming the pistons are together at the centre of the cylinder, the explosion takes place and the pistons move apart. At a certain moment the front piston uncovers the ring of exhaust ports *E*, and the pressure falls to about that of the atmosphere. The rear piston now uncovers the ring of air ports *A*, and the scavenging air is admitted. A little later the gas ports *G* are uncovered and the "mixture" is formed. On the return stroke the ports are closed in the same order (exhaust, gas, air), and the mixture is then compressed between the approaching pistons, and fired when they are again together. The length of the pistons is such that the ports remain closed in this position.

The gas and air pumps may be arranged in one of three ways, viz.—

	Position of Pumps	Method of Driving
(*a*)	Tandem with main cylinder	Off rear crosshead
(*b*)	Side of main cylinder	Off separate crank
(*c*)	Below floor level	Through rocking lever

This engine is, of course, of great length, and is comparatively costly to manufacture, but it has several advantages, including simplicity, accessibility, and good balance.

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
Lecturer in Mathematics, Battersea Polytechnic

LESSON XIII

GEOMETRY—(*contd.*)

Angles of Polygons. Suppose a man to walk round a field in shape like $ABCDE$ (Fig. 31). At A he will turn through the exterior angle, $\angle a$, between AB and EA produced, and similarly at the other corners. In a complete circuit, he will have made one complete turn, that is, he will in all have turned through 4 right angles or 360°. Therefore (since he does *no* turning in walking along the *straight* sides),

$$\angle a + \angle b + \angle c + \angle d + \angle e = 360^\circ.$$

The same reasoning will hold for *any* polygon, so that **the sum of the exterior angles of any polygon is 360°.**

If the polygon has n sides, and consequently n angles, then, since the sum of the exterior and interior angles at any corner is 180°, the sum of all the exterior and all the interior angles will be $n \times 180^\circ = 180n^\circ$. Now we have just seen that the sum of the *exterior* angles is 360°, so that the sum of the *interior* angles must be

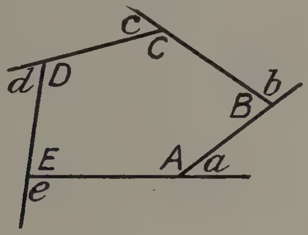


FIG. 31

$$(180n - 360)^\circ \quad \text{or} \quad 2n - 4 \text{ right angles.}$$

Thus, in a pentagon (5 sides) the sum of the angles is $(2 \times 5) - 4 = 6$ right angles, or 540°. In a regular pentagon, when all the angles are equal, each must be $540^\circ \div 5 = 108^\circ$, and the angles of other regular polygons may be similarly determined.

Triangles. Imagine a framework of four rods, $ABCD$ (Fig. 32), pivoted at the joints. It is possible to alter its shape, to such as ABC_1D_1, or in other ways, without undoing the joints. Such a framework is not, of itself, rigid. There are, however, several ways in which it can be made rigid. One is by means of another bar from A to C (or, of course, from B to D). This, in effect, divides the figure into two triangles, and we know that a triangular framework *is* rigid. This means that if we have the three sides of a triangle, it must be of a fixed shape; two such made with bars of the same respective lengths are exactly equal—angles must be equal too. Another way of putting the same fact is to say that to copy a given triangle, we need only copy the sides. If we make their lengths the same in the copy as in the original, the angles of the copy *must* be the same too. Returning to our framework of four bars, another way of making it rigid is to clamp the joint at B, that is, in effect, to fix the angle B. This, again, fixes the *triangle* ABC, and, since it keeps A and C at a fixed distance apart, it also fixes the triangle DAC. The same fact, that two sides and the angle between them fix a triangle, can be approached by thinking of copying a triangle, when, if these are correctly copied, the other two angles and the other side *must* come right. Another way of copying is to copy one side, and the angles at its ends. If this is correctly done, the other angle and the other two sides *must* come right. Thus, if we wish to copy a triangle, or to test

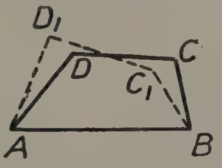


FIG. 32

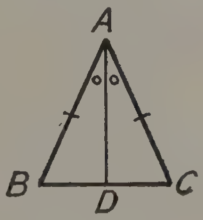


FIG. 33

whether two triangles are equal, we need only copy (or test)—

1. **Three sides,** or
2. **Two sides and the included angle,** or
3. **One side and two angles.**

In (2) it must be the *included* angle—see for yourself what might happen if you choose one of the other angles instead. In (3), *any* two angles will do, since the sum of all three is 180°, so that, given two, we can find the third.

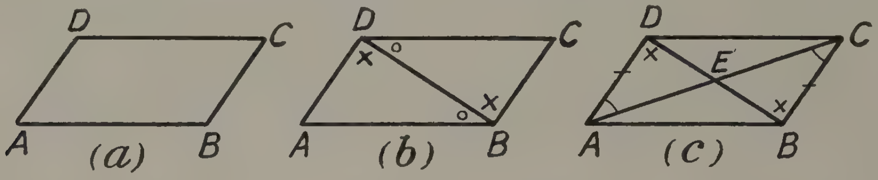


FIG. 34

Isosceles Triangles. If a triangle has two sides equal, it is called an isosceles triangle. Such a triangle must also have two angles equal. This is almost obvious, but can be proved by drawing a line to bisect the vertical angle. This makes two triangles (see Fig. 33), which are equal by (2) above, and consequently $\angle B = \angle C$. If a triangle has all three sides equal, it is called equilateral, and must also have all three angles equal. Consequently, each must be 60°.

Parallelogram. A parallelogram is a four-sided figure (quadrilateral) with two pairs of parallel sides (Fig. 34*a*). It also must have opposite sides and angles *equal.* Join DB (Fig. 34*b*). Then, by the properties of parallels, the angles similarly marked are equal. Consequently, the triangles ABD and CDB, having a common side BD, and these angles equal, must be equal. Therefore, $AB = DC$, $AD = BC$, $\angle A = \angle C$; also, evidently, $\angle B = \angle D$. There is also another fact of importance; the diagonals bisect one another. In Fig. 34*c*, marked sides and angles are equal, for reasons already given, so that triangle ADE = triangle CBE. Therefore, $AE = EC$ and $BE = ED$.

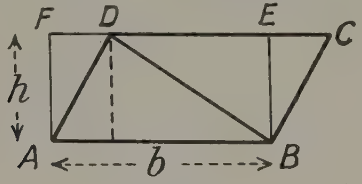


FIG. 35

EXERCISE No. 22

1. Show by the polygon rule that the sum of the angles of a triangle must be 180°.
2. In roughly surveying a field the shape of which is a pentagon, the angles found were 107°, 96°, 144°, 73°, and 125°. What was the total error? If the error is equally distributed among the angles, what are their correct values?
3. Find the angle of a regular nonagon (9 sides).
4. The angle of a regular polygon is 162°. How many sides has it?
5. In a triangle ABC, $\angle B = \angle C$. Show that AB must be equal to AC. (Hint: draw AD perpendicular to BC.
6. In the quadrilateral $ABCD$, it is known that $\angle A = \angle B$, and that $AD = BC$. Show that $\angle C$ must be equal to $\angle D$. (Hint: produce AD and BC to meet in E.)
7. Two lines AB, CD, are equal and parallel. Show that AC and BD must be equal and parallel.

Areas. The area of a rectangle is found by multiplying length by breadth. For a parallelogram, we have only to replace breadth by the *perpendicular* distance between two sides. For, in Fig. 35, in which $ABCD$ is a parallelogram, and where BE and AF are perpendicular to AB, the triangles BEC and AFD are equal. We may imagine the triangle BEC cut off and placed at AFD, when we see that the area of the parallelogram $ABCD$ must be equal to the area of the rectangle $AFEB$, i.e. area of parallelogram $= bh$.

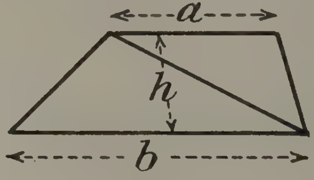


FIG. 36

We already know that the diagonal BD bisects the parallelogram, so that the area of the triangle ABD is half that of the parallelogram, i.e. is $\frac{1}{2}bh$. Now any two equal triangles can be placed together to make a parallelogram, so that for *any* triangle, area $= \frac{1}{2}$ (base) × (perpendicular height).

A trapezium, that is, a quadrilateral with two parallel sides (see Fig. 36), can be divided into two triangles, with the *same* height. Thus, in the figure, the areas of the two triangles are

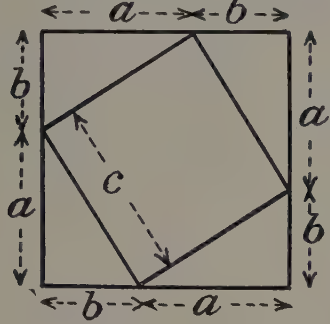


FIG. 37

$\frac{1}{2}ah$ and $\frac{1}{2}bh$ respectively, so that the area of the trapezium $= \frac{1}{2}ah + \frac{1}{2}bh = \frac{1}{2}h\,(a + b)$. Another way of putting this is $\frac{a + b}{2}\,h$, i.e. (average length) × (height). Any polygon can be divided into triangles and trapezia, so that its area can be found by drawing and measurement.

Right-angled Triangles. Theorem of Pythagoras. We now come to what is *quite* the most important fact in elementary geometry, namely that, given two sides of a right-angled triangle, it is possible to *calculate* the third. It must have been obvious to the users of tiles whose shape is obtained by halving a square diagonally, that the square on the diagonal is twice the area (4 tiles) of the square (2 tiles)—that the square

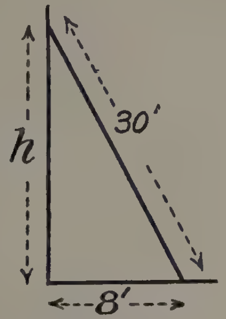


FIG. 38

on the longest side of the tile is the sum of the squares on the other two sides. That a triangle with sides 3, 4, and 5 units long is right angled was known to the Egyptians who built the pyramids.

To see that if the triangle is right angled, the sum of the squares on two sides is equal to the square on the other (as $3^2 + 4^2 = 5^2$), let the sides be a, b, and c, and by using four equal

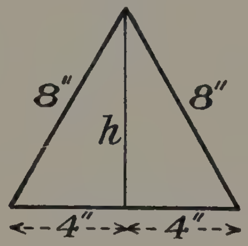


FIG. 39

triangles build up a square as in Fig. 37. In that figure,

inner square = outer square − four triangles

$$c^2 = (a + b)^2 - (4 \times \tfrac{1}{2}ab)$$
$$= a^2 + 2ab + b^2 - 2ab$$

so that

$$\underline{c^2 = a^2 + b^2.}$$

EXAMPLE 1. A ladder 30 ft. long has its foot 8 ft. from the foot of a wall. How far up the wall will it reach?

This is represented diagrammatically in Fig. 38. Using the above relation between the sides of the right-angled triangle, calling the required height h,

$$30^2 = 8^2 + h^2$$

whence $h^2 = 30^2 - 8^2 = 900 - 64 = 836$

so that $h = \sqrt{836} = \underline{28{\cdot}9 \text{ ft. (approx.)}}$.

EXAMPLE 2. Find the height and area of an equilateral triangle with sides 8 in. long.

In Fig. 39 we have drawn the perpendicular, which bisects the base also, and thus divides

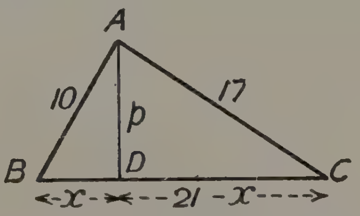


FIG. 40

the triangle into two equal right-angled triangles. Using our rule for one of these

$$8^2 = 4^2 + h^2$$

whence $h^2 = 8^2 - 4^2 = 64 - 16 = 48$

so that $h = \sqrt{48} = \underline{6{\cdot}928 \text{ in.}}$

Consequently, area $= \frac{1}{2} \times 8 \times 6{\cdot}928$

$= 27{\cdot}71 \ldots$ sq. in.

EXAMPLE 3. The sides of a triangle are 21, 17, and 10 ft. Find its area.

In the sketch (Fig. 40), we have used the 21 ft. side as base, and drawn the perpendicular AD. This divides the triangle into two right-angled triangles. As, however, the triangle is not isosceles, the foot of the perpendicular will not bisect the base. We have, then, to find the segments of the base as well as the height. Call DB x, when DC will be $21 - x$; also call AD p. Then, by the rule, we have

for triangle ABD, $p^2 = 10^2 - x^2$

for triangle CBD, $p^2 = 17^2 - (21 - x)^2$

equating these two values of p^2, and working out,

$$100 - x^2 = 289 - (441 - 42x + x^2)$$
$$= 289 - 441 + 42x - x^2.$$

Transposing, the x^2 terms disappear, and we get

$$100 - 289 + 441 = 42x$$

or $42x = 252$

whence $x = 6.$

Now using this value of x in either equation for p^2 (both should give the same; their doing so is a check upon our work) we find

$p^2 = 64$, whence $p = 8$ ft.

The area is, now,

$\frac{1}{2} \times 21 \times 8 = 84$ sq. ft.

EXERCISE No. 23

1. By drawing and measurement, find the areas of (*a*) an isosceles triangle with sides, 2 in., 2 in., and 3 in.; (*b*) a regular hexagon with side, 2 in.; (*c*) a triangle with sides 7·5, 7·0, and 6·5 cm.

2. A wireless mast 30 ft. high has guy ropes 35 ft. long. How far from the foot of the mast are they fixed?

3. A right-angled triangle has the sides including the right angle 6·6 and 11·2 cm. long. Find the length of the hypotenuse, and also the perpendicular from the right-angled vertex to the hypotenuse.

4. Find, by calculation, the areas of the three figures of Example (1) above.

ANSWERS TO EXERCISES NOS. 20 AND 21

(20)

1. 2 or 4.
2. -1 or -3.
3. 1·646 . . . or $-3{\cdot}646$ . . .
4. 2·580 . . . or 0·620 . . .
5. $\frac{1}{6}$ or -1.
6. 3 or $-\frac{1}{3}$.
7. 0·211 . . . or 0·789 . . .
8. 3 or 5.
9. 12 or -13.
10. 1 or $-1{\cdot}5$.
11. 1·387 . . . or $-1{\cdot}262$ . . .
12. 17·04 . . . or $-39{\cdot}04$ . . .

(21)

1. $x^5 + 5x^4a + 10x^3a^2 + 10x^2a^3 + 5xa^4 + a^5$.
2. $1 + 10x + 45x^2 + 120x^3 + 210x^4 + 252x^5 + 210x^6 + 120x^7 + 45x^8 + 10x^9 + x^{10}$.
3. 1·1041
4. 6·33 . . ., 9·85 . . ., 3·11 . . .

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XIII

DEFLECTION OF BEAMS DUE TO BENDING

The total deflection at any point in a loaded beam is due partly to flexure and partly to shear strain, but with beams of ordinary proportions the deflection due to shear is negligible in comparison with that due to bending. The simplest case of deflection, due to bending, occurs when the beam is subjected to a constant B.M.

$$\text{Since } \frac{M}{I} = \frac{E}{R} \quad \therefore \quad R = \frac{EI}{M}$$

The beam being homogeneous and of constant cross-section, it will bend into a circular arc, that is, a curve of constant radius.

Circular bending is also produced where the I of the beam section varies with the applied B.M., so that the ratio $\frac{I}{M}$ is a constant.

Deflection due to Bending into a Circular Arc. Consider a beam which is subjected to a B.M. so that it bends into a circular arc of radius R, as shown in Fig. 66. From the geometry of the circle we know that

$$(2R - \triangle)\triangle = \left(\frac{l}{2}\right)^2,$$

$$\text{or} \quad 2R\triangle - \triangle^2 = \frac{l^2}{4} \qquad . \quad . \quad (1)$$

where $\triangle$ = deflection at the centre measured below the supports.

But, under ordinary circumstances, $\triangle$ will be small, and, therefore, $\triangle^2$ very small compared to R, so we may neglect Δ^2 in (1) and write the expression

$$2R\triangle = \frac{l^2}{4}$$

$$\text{or} \quad \triangle = \frac{l^2}{8R} \qquad . \quad . \quad . \quad (2)$$

Carriage Springs

It is usual to apply equation (2) in calculating the deflection of a carriage or laminated spring, although friction and other considerations of manufacture modify the result.

The shape of the spring is such, that, approximately, the moment of inertia of the cross-section varies at the same rate as the bending moment, from which it follows that $\frac{I}{M}$ is constant, and, E being constant

$$\therefore \text{ since } R = \frac{EI}{M}$$

or R = a constant, and the beam bends into the arc of a circle.

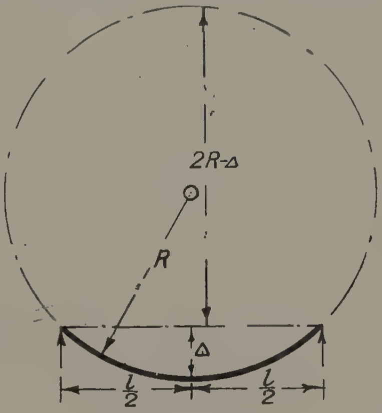


Fig. 66

Example. A carriage spring, 25 in. long and 2 in. wide. is required to straighten under a central load of 2,000 lb. If the plates forming the spring are $\frac{1}{2}$ in. thick, calculate the number of such plates required if the stress is not to exceed 30,000 lb. per sq. in. Calculate the initial radius of curvature, and the central deflection under this load. $E = 30 \times 10^6$ lb. per sq. in.

I (for one plate) $= \frac{2}{3}bd^3$
" " $= \frac{2}{3} \times 2 \times \frac{1}{64}$
" " $= \frac{1}{48}$
b = breadth = 2 in.
d = half thickness = $\frac{1}{4}$ in.
n = number of plates

$$\therefore \text{ total } I = \tfrac{1}{48} \times n$$

$$\text{Also } \frac{M}{I} = \frac{f}{y}, \quad \text{and } M = \frac{Wl}{4}$$

$\therefore \frac{Wl}{4I} = \frac{f}{y}$; and substituting the known values, we get, $\frac{2000 \times 25 \times 48}{4 \times n} = \frac{30{,}000 \times 4}{1}$

$\therefore n = 5$ plates.

Again, $\frac{f}{y} = \frac{E}{R}$ $\therefore R = \frac{Ey}{f} = \frac{30{,}000{,}000 \times \frac{1}{4}}{30{,}000}$

or $R = 250$ in.

Also, $\triangle = \frac{l^2}{8R} = \frac{25 \times 25}{8 \times 250} = \underline{0{\cdot}3125 \text{ in. deflection.}}$

Example. The axle of a railway truck is 3 in. in diameter; the axle boxes and wheels are symmetrically placed at centre distances of 5 ft. 8½ in. and 4 ft. 8½ in. respectively. Determine the upward deflection of the centre of the axle, when each axle box is loaded with 2 tons, as shown in Fig. 66A. Take $E = 13{,}000$ tons per sq. in.

It will be seen from the B.M. diagram that the part of the axle between the wheels is acted upon by a *constant* bending moment of $2 \times 6 = 12$ tons-in., and hence bends into the arc of a circle. The radius of this circular arc

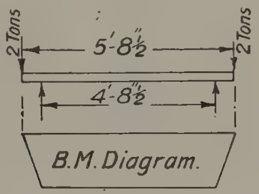


Fig. 66A

$$= R = \frac{EI}{M} = \frac{13{,}000 \times \pi \times (3)^4}{12 \times 64} = 4307 \text{ in.}$$

$$\text{Also } \triangle = \frac{l_2}{8R} = \frac{56{\cdot}5 \times 56{\cdot}5}{8 \times 4307} = \underline{0{\cdot}09264 \text{ in.}}$$

DEFLECTION OF CANTILEVER WITH END LOAD

We have shown in Lesson IX that

$$\frac{1}{R} = \frac{M}{EI} = \frac{d^2y}{dx^2}$$

(a very close approximation),

$$\text{or } M = EI \cdot \frac{d^2y}{dx^2};$$

and we shall make frequent use of this expression in determining the deflection of beams.

Consider any section AA, of the loaded cantilever shown in Fig. 67. The B.M. on section $AA = W(l - x)$.

$$\therefore M = EI \cdot \frac{d^2y}{dx^2} = W(l - x) \quad . \quad . \quad (1)$$

Integrating this we get the slope equation

$$EI. \frac{dy}{dx} = W\left(lx - \frac{x^2}{2}\right) + C \quad . \quad . \quad (2)$$

To determine the value of the constant C, we note that when $x = 0$ (at B, see Fig. 67), $\frac{dy}{dx} = 0$; that is, the beam is horizontal at entrance to the wall or other rigid end fixing.

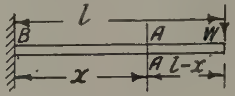


Fig. 67

Substituting $x = 0$ and $\frac{dy}{dx} = 0$ in (2) we see that C also $= 0$. Integrating (2) we obtain the deflection (y) equation—

$$EIy = W\left(\frac{lx^2}{2} - \frac{x^3}{6}\right) + D \quad . \quad . \quad (3)$$

Again, we have to determine the value of the constant D, and for this purpose we note that when $x = 0$ (at B), $y = 0$. Substituting these values in (3) we see that $D = 0$, hence

$$y = \frac{W}{EI}\left(\frac{lx^2}{2} - \frac{x^3}{6}\right) \quad . \quad . \quad . \quad (4)$$

an equation which can be used to determine the deflection at any beam section whose distance, x, from the origin B is known.

The maximum deflection occurs under the load, that is, where $x = l$; substituting this in (4) we get—

$$y = \frac{W}{EI}\left[\frac{l^3}{2} - \frac{l^3}{6}\right] = \underline{\frac{Wl^3}{3EI}}$$

DEFLECTION OF CANTILEVER WITH AN EVENLY DISTRIBUTED LOAD

Let the load per unit length be denoted by w_3. Then the total load $W = wl$.

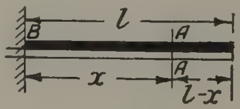


Fig. 68

At any section AA, the B.M.

$$= M = EI\frac{d^2y}{dx^2} = w(l - x)\left(\frac{l - x}{2}\right)$$

$$M = EI\frac{d^2y}{dx^2} = \frac{w}{2}(l^2 - 2lx + x^2) \quad . \quad . \quad (5)$$

Integrating (5) we get—

$$EI\,\frac{dy}{dx} = \frac{w}{2}\left(l^2x - \frac{2lx^2}{2} + \frac{x^3}{3}\right) + F \quad . \quad (6)$$

But, when $x = 0$, $\frac{dy}{dx} = 0 \quad \therefore \; F = 0.$

Integrating (6) we get—

$$EIy = \frac{w}{2}\left(\frac{l^2x^2}{2} - \frac{lx^3}{3} + \frac{x^4}{12}\right) + G \quad . \quad (7)$$

Again when $x = 0$, $y = 0 \;\therefore\; G = 0$ and (7) may be written

$$y = \frac{w}{2EI}\left(\frac{l^2x^2}{2} - \frac{lx^3}{3} + \frac{x^4}{12}\right) \quad . \quad . \quad (8)$$

The maximum deflection occurs at the free end where $x = l$, and is equal to

$$y = \frac{w}{2EI}\left(\frac{l^4}{2} - \frac{l^4}{3} + \frac{l^4}{12}\right) = \frac{wl^4}{8EI} = \underline{\frac{Wl^3}{8EI}}$$

Deflection of Beam Supported at its Ends and Loaded at Mid-span (Fig. 69)

In this case we shall take the origin at mid-span. At any section AA the B.M.

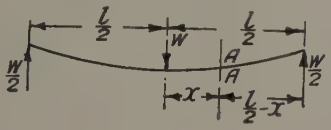


Fig. 69

$$= M = EI\,\frac{d^2y}{dx^2} = \frac{W}{2}\left(\frac{l}{2} - x\right) \quad . \quad . \quad (9)$$

Integrating (9) we get

$$EI\,\frac{dy}{dx} = \frac{W}{2}\left(\frac{lx}{2} - \frac{x^2}{2}\right) + K \quad . \quad . \quad (10)$$

and when $x = 0$, $\frac{dy}{dx} = 0 \quad \therefore K = 0.$

Integrating (10) we get

$$EIy = \frac{W}{2}\left(\frac{lx^2}{4} - \frac{x^3}{6}\right) + L \quad . \quad . \quad (11)$$

and when $x = 0, y = 0 \quad \therefore \; L = 0.$

The maximum deflection is upward and occurs at the supports where $x = \frac{l}{2}$; substituting this in equation (11) we get

$$y = \frac{W}{2EI}\left(\frac{l^3}{16} - \frac{l^3}{48}\right) = \underline{\frac{Wl^3}{48EI}}$$

Deflection of Beam Supported at its Ends and Loaded Uniformly

Again we take the origin at the mid-span and measure the deflection upwards.

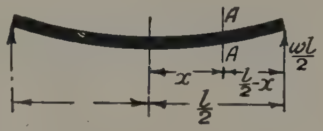


Fig. 70

At any section AA, the B.M. (Fig. 70)

$$= M = EI\,\frac{d^2y}{dx^2} = \frac{wl}{2}\left(\frac{l}{2} - x\right)$$

$$-\frac{w}{2}\left(\frac{l}{2} - x\right)\left(\frac{l}{2} - x\right)$$

$$\text{or } M = EI\,\frac{d^2y}{dx^2} = \frac{w}{2}\left(\frac{l^2}{4} - x^2\right) \quad . \quad . \quad (12)$$

Integrating, we get

$$EI\,\frac{dy}{dx} = \frac{w}{2}\left(\frac{l^2x}{4} - \frac{x^3}{3}\right) + P \quad . \quad . \quad (13)$$

But $\frac{dy}{dx} = 0$ when $x = 0. \quad \therefore \; P = 0.$

Integrating again, we have

$$EIy = \frac{w}{2}\left(\frac{l^2x^2}{8} - \frac{x^4}{12}\right) + Q \quad . \quad . \quad (14)$$

When $x = 0, y = 0. \quad \therefore \; Q = 0.$

The maximum deflection is upwards and occurs at the supports where $x = \frac{l}{2}$; substituting in (14), we have

$$\therefore \; y = \frac{w}{2EI}\left(\frac{l^4}{32} - \frac{l^4}{192}\right) = \frac{5wl^4}{384EI}$$

$$= \underline{\frac{5Wl^3}{384\,EI}} \quad (15)$$

Deflection of Beam Rigidly Fixed at its Ends and Loaded at Mid-span

At any section AA the B.M. (Fig. 71)

$$= M = EI\,\frac{d^2y}{dx^2} = \frac{W}{2}\left(\frac{l}{2} - x\right) - M_R \quad (16)$$

where M_R = the fixing moment acting about the right-hand support.

Integrating (16) we get

$$EI\frac{dy}{dx} = \frac{W}{2}\left(\frac{lx}{2} - \frac{x^2}{2}\right) - M_R x + S \quad . \quad (17)$$

when $x = 0$, $\frac{dy}{dx} = 0$. $\therefore$ $S = 0$.

Integrating (17) we get

$$EIy = \frac{W}{2}\left(\frac{lx^2}{4} - \frac{x^3}{6}\right) - \frac{M_R x^2}{2} + T \quad . \quad (18)$$

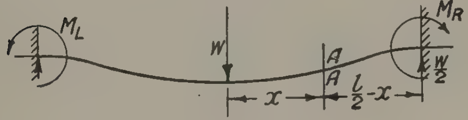


FIG. 71

Again when $x = 0$, $y = 0$. $\therefore$ $T = 0$.
To evaluate the end fixing moment M_R, we use the relation $\frac{dy}{dx} = 0$ when $x = \frac{l}{2}$ in equation (17) and get

$$0 = \frac{W}{2}\left(\frac{l^2}{4} - \frac{l^2}{8}\right) - M_R \cdot \frac{l}{2}$$

$$\therefore M_R = \frac{Wl}{8}$$

This value of M_R can now be substituted in equation (18) and the maximum deflection, which occurs at the supports where $x = \frac{l}{2}$, found as follows—

$$y = \frac{W}{2EI}\left(\frac{l^3}{16} - \frac{l^3}{48}\right) - \frac{Wl}{8} \times \frac{l^2}{8}$$

$$\therefore \quad y = \frac{Wl^3}{192EI}$$

DEFLECTION OF BEAM RIGIDLY FIXED AT ITS ENDS AND LOADED UNIFORMLY

At any section AA, the B.M. (Fig. 72)

$$= M = EI \cdot \frac{d^2y}{dx^2} = \frac{wl}{2}\left(\frac{l}{2} - x\right)$$

$$- \frac{w}{2}\left(\frac{l}{2} - x\right)\left(\frac{l}{2} - x\right) - M_R$$

$$\text{or } M = EI \cdot \frac{d^2y}{dx^2} = \frac{wl^2}{8} - \frac{wx^2}{2} - M_R \quad . \quad (19)$$

Integrating, we get

$$EI \cdot \frac{dy}{dx} = \frac{wl^2x}{8} - \frac{wx^3}{6} - M_R x + R \quad . \quad (20)$$

And since $\frac{dy}{dx} = 0$, when $x = 0$. $\therefore$ $R = 0$.

Integrating (20), we get

$$EI \cdot y = \frac{wl^2x^2}{16} - \frac{wx^4}{24} - M_R \cdot \frac{x^2}{2} + S \quad . \quad (21)$$

Again $y = 0$ when $x = 0$. $\therefore$ $S = 0$.
To find the value of the end fixing moment, we note that, when $x = \frac{l}{2}$, $\frac{dy}{dx} = 0$, and

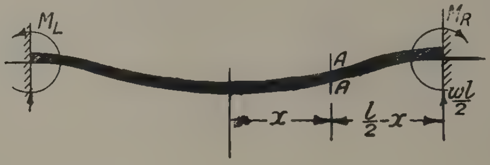


FIG. 72

substituting these values in (20), we get

$$0 = \frac{wl^3}{16} - \frac{wl^3}{48} - M_R \cdot \frac{l}{2}$$

$$\therefore M_R = \frac{wl^2}{12} \quad . \quad . \quad (22)$$

Substituting the value of M_R found above in equation (21) and $x = \frac{l}{2}$ to obtain the maximum value of the deflection which occurs at the supports, we have

$$y = \frac{1}{EI}\left[\frac{wl^4}{64} - \frac{wl^4}{384} - \frac{wl^4}{96}\right]$$

$$\therefore y = \frac{Wl^3}{384EI}$$

The above results are collected and tabulated as under—

Case	Type of Beam and Loading	Maximum B.M.	Maximum Deflection
1	Cantilever, load at free end	Wl	$\frac{Wl^3}{384EI} \times 128$
2	Cantilever, load uniformly distributed	$\frac{Wl}{2}$	$\frac{Wl^3}{384EI} \times 48$
3	Simply supported with central load	$\frac{Wl}{4}$	$\frac{WL^3}{384EI} \times 8$
4	Simply supported uniformly distributed load	$\frac{Wl}{8}$	$\frac{Wl^3}{384EI} \times 5$
5	Fixed ends with central load	$\frac{Wl}{8}$	$\frac{Wl^3}{384EI} \times 2$
6	Fixed ends with uniformly distributed load	$\frac{Wl}{12}$	$\frac{Wl^3}{384EI} \times 1$

THE THEORY OF HEAT ENGINES

By E. B. Cole, B.Sc.

LESSON IX

ENTROPY DIAGRAMS

T - ϕ chart for a liquid and its vapour under different constant pressures.

In the last lesson we followed on the $T-\phi$ diagram the heating of 1 lb. of water stuff under a constant pressure, and the student should now be able to understand Fig. 19, which represents a series of these curves for a range of constant pressures above and below the critical. The diagram is very much exaggerated for the sake of clearly understanding the ideas involved.

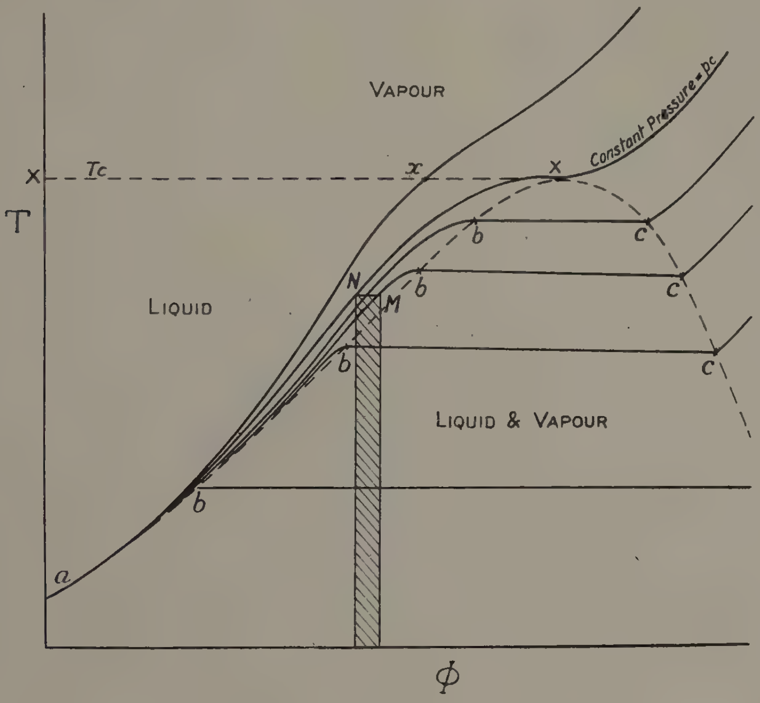


Fig. 19.

The curves ab represent the warming of the liquid, at b we have arrived at the temperature at which the vapour pressure of the liquid is equal to the external pressure, and evaporation occurs. At c the liquid has all been converted into dry saturated vapour and superheating begins.

By joining the points bbb we get a boundary curve called the *liquid line*, while the curve joining points ccc is called the *vapour line*. The junction of these two curves give the critical point X; heating under a constant pressure equal to or higher than the critical pressure p_c results in the liquid changing its state to vapour without any horizontal break in the curve, since no latent heat is required. Hence, if XX represents the critical temperature line, change of state takes place at x, and in the space above XX and to the right of boundary cc, the substance can only exist as a vapour, while below XX and to the left of boundary bb liquid can only exist, and within bb and cc liquid and vapour exist together.

Now we have seen that at temperatures far removed from and below the critical a liquid can be considered incompressible, becoming more and more compressible as we approach the critical temperature. Thus, if we can compress a liquid, whose condition is represented initially by the point M on the constant pressure curve p_1, from the pressure p_1 to the pressure p_2, say, at constant temperature, then heat will be rejected and the horizontal line MN represents the loss of entropy that has occurred, where N is a point at the same temperature as M but on the constant pressure curve p_2. The amount of heat rejected will be given by the area under MN.

As we recede from the critical temperature and the liquid becomes less compressible, i.e. there is a smaller change in volume on compressing over the same range at this lower constant temperature and, therefore, less work is done, the constant pressure curves approach and finally merge into one another, becoming indistinguishable from the boundary curve. For all ordinary temperature ranges employed in steam work water may be considered incompressible, so that we shall be working on the lower portion of the diagram in which the "liquid line" will represent the warming of water under any constant pressure.

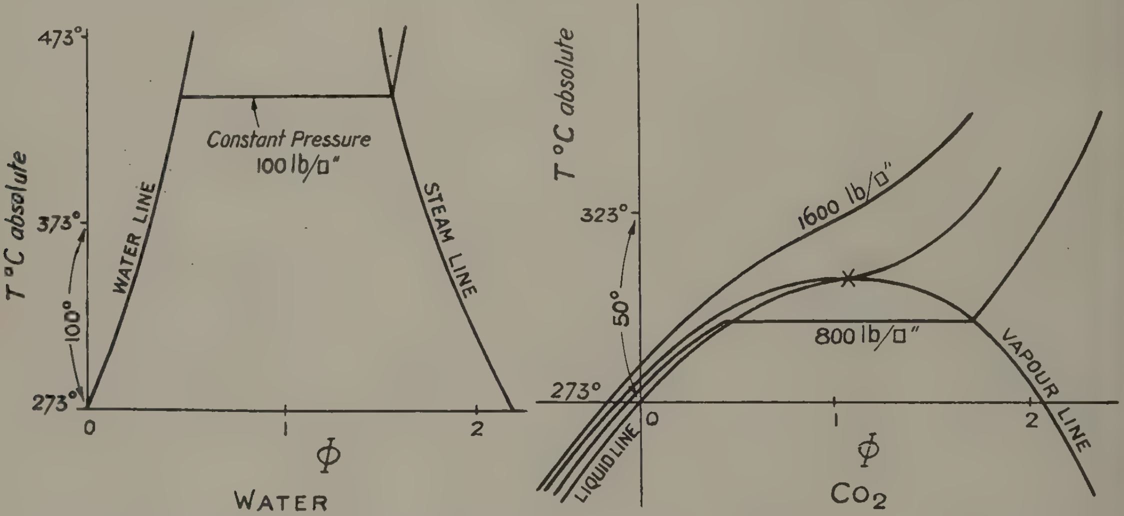


FIG. 20. T-ϕ CHARTS FOR WATER AND CO_2

For refrigeration work with CO_2, the portion of the $T-\phi$ chart for that substance round about the critical temperature is of the most importance, and for the sake of comparison the charts for water and CO_2 are sketched approximately to scale for the same range of temperature in Fig. 20.

Constant Volume Lines on the T – ϕ Chart. Consider any evaporation line, such as bc, on the $T-\phi$ chart, Fig. 21. If only $\frac{1}{x}$th of a pound of water has been evaporated the state point will be at f, such that

$$bf/bc = \frac{1}{x}$$

The change in entropy, in other words, is only $\frac{1}{x}$ of that which is produced during the complete evaporation of the pound of water into dry steam and, neglecting the small volume occupied by the water, the volume of steam produced will be $\frac{1}{x}$ of that of 1 lb. dry saturated steam.

Hence, if such lines as bc be taken to represent to the same scale the volume of 1 lb. dry and saturated steam at that temperature, then they may be graduated accordingly—and lines drawn through corresponding points representing equal volumes of steam formed, are termed *constant volume lines.* The figure explains the idea.

Constant Quality Lines. Another set of curves will be found on a complete $T-\phi$ chart, namely curves representing constant dryness fraction of the steam. These are obviously obtained by dividing the evaporation lines bc at points ff, such that in each case the dryness fraction bf/bc is constant and joining up these points (Fig. 22).

ADIABATIC EXPANSION OF STEAM

We have defined adiabatic expansion as that in which work is done by the substance, but no heat is received from or rejected to an external source—the work done, therefore, being equal to the difference between the initial and final internal energies.

The expansion curve on the $T - \phi$ diagram will be represented by a vertical straight line such as fk (Fig. 23), where the point f represents the initial state of 1 lb. of steam and k the final condition. Three such curves for steam are represented in which the steam is initially (*a*) wet, (*b*) dry, and (*c*) superheated, and it is required to determine the final quality of the steam after the expansion. This can, of course, be determined by direct measurement from the chart: dryness fraction $= ak/am$. It will be noticed that if the steam is initially dry the steam becomes wetter on expansion.

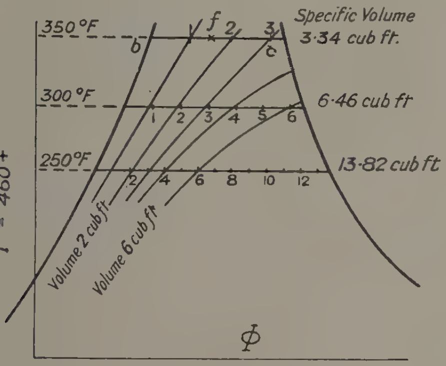


FIG. 21. CONSTANT VOLUME LINES

Similarly so, if the steam is fairly dry at the start, but, if it is very wet, such as at f^1 initially, it will be found that it becomes drier after expansion, i.e. $\frac{ak^1}{am} > \frac{bf^1}{bc}$.

With superheated steam, the superheat is gradually reduced until at r the steam is just dry and saturated. Expansion below this point results, of course, in the steam becoming wet.

Case (*a*). Dryness fraction x_2 after adiabatic expansion from an initial absolute temperature T_1 and dryness x_1 to temperature T_2.

$$x_2 = \frac{ak}{am} = \frac{ae + ek}{am}$$

But

$$ae = \log_e \frac{T_1}{T_2} \text{ (Here } T_1 > T_2\text{)}$$

$$ek = bf = x_1 \cdot \frac{L_1}{T_1}$$

$$am = L_2/T_2$$

$$\therefore \quad x_2 = \frac{\log_e \frac{T_1}{T_2} + x_1 \cdot \frac{L_1}{T_1}}{L_2/T_2}$$

Case (*b*). Similar to above, but $x_1 = 1$.

Case (*c*). On the assumption that the expansion is carried so far that wet steam results

$$ak = ae + es + k_p \log_e \cdot \frac{T_s}{T_1}$$

$$\therefore \quad x_2 = \frac{\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + k_p \log_e \frac{T_s}{T_1}}{L_2/T_2}$$

Let us now see how the work done during expansion may be computed.

Method 1. Assuming that the expansion curve on the PV diagram is of the general type $PV^n = C$, it is then necessary to determine the value of n.

For a gas we have seen that for adiabatic expansion $n =$ ratio of K_p/K_v, i.e. γ, but it will be clear that this cannot apply to a substance whose quality is varying.

The value of n that will fit the expansion curve over different ranges of pressure will probably be a variable, but a mean value which will approximate over the whole range may be found as follows—

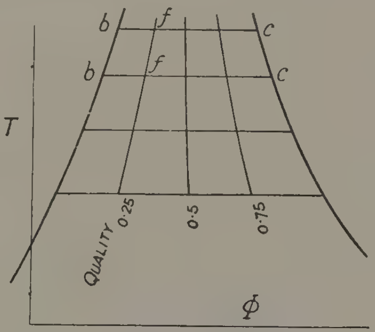


FIG. 22. CONSTANT QUALITY LINES

If $PV^n = C$, then taking logs.

Log. $P + n$ log. $V =$ log. C,

which is the equation of a straight line, and n is the slope of the line.

Thus, if a series of values of P be taken over the range of the expansion, and a corresponding series of values of V determined—then plotting log. P against log. V a straight line should result.

Actually, as the value of n that will fit the curve varies over different portions, the plotted points will not lie quite on a straight line, but by drawing the best straight line through them a mean value of n over the whole range may be obtained.

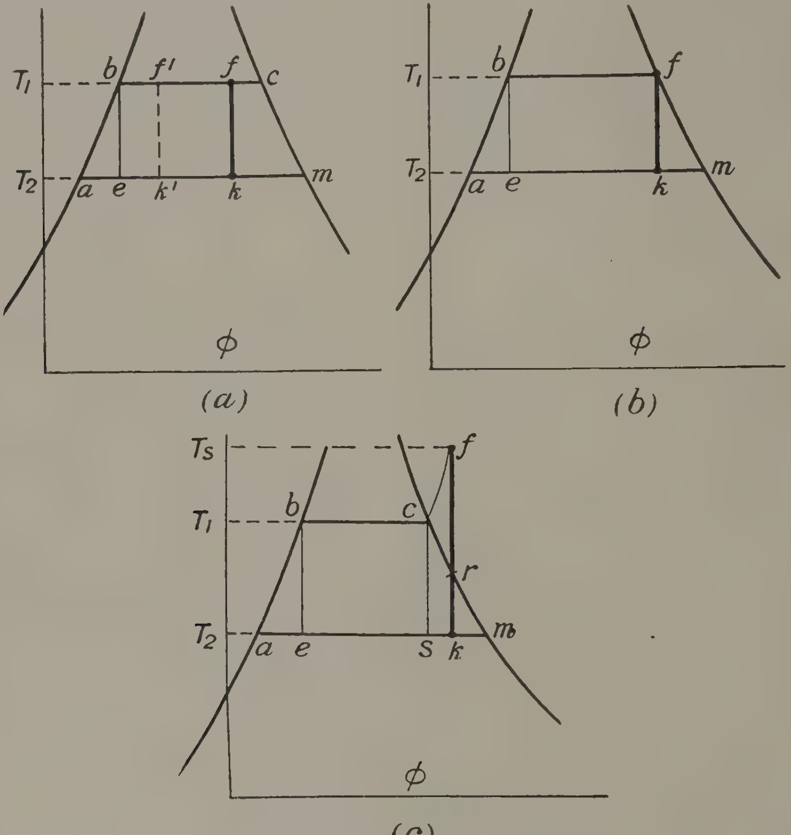


FIG. 23. ADIABATIC EXPANSION

To find the value of V_2 at any intermediate pressure P_2, determine the dryness fraction x_2 after expansion down to P_2 as above, then, if V_{s2} = volume of 1 lb. of dry steam at this pressure, the actual volume is $x_2 V_{s2}$. Similarly for the volumes corresponding to the remaining pressures taken.

The work done over the range 1 – 2 is then

$$\frac{P_1V_1 - P_2V_2}{n - 1} \text{ ft.-lb.}$$

(It may be as well to remind the student that $PV = wRT$ only applies to a *gas* and cannot be substituted here.)

Method 2. The work done is equal to the change in internal energy of the steam.

The dryness fraction x_2 at the end of the expansion must again be determined from the $T - \phi$ chart.

Then the initial internal energy (above that of water at 32° F.) of the steam is given by: sensible heat + x (internal latent heat), i.e. $h_1 + x_1\rho_1$ (see Lesson VI).

Similarly after expansion the internal energy is $h_2 + x_2\rho_2$.

These values per pound of steam are obtainable from the tables, and the work done is then $h_1 - h_2 + x_1\rho_1 - x_2\rho_2$ heat units per lb.

EXAMPLE. Determine the work done per pound during the adiabatic expansion of steam initially of dryness 0·97 and pressure 200 lb.-sq. in. abs. to a pressure of 1 lb. per sq. in.

To determine x_2

From steam tables we have

$$T_1 = (460 + 381{\cdot}9)^\circ \text{ F. abs.}$$
$$L_1 = 843{\cdot}2 \text{ B.Th.U.}$$
$$T_2 = (460 + 101{\cdot}83)$$
$$L_2 = 1034{\cdot}6.$$

Then substituting in dryness fraction equation

$$x_2 = 0{\cdot}751$$

Again from tables

$$h_1 = 354{\cdot}9 \text{ B.Th.U.}$$
$$h_2 = 69{\cdot}8$$
$$\rho_1 = 759{\cdot}5$$
$$\rho_2 = 972{\cdot}9.$$

$\therefore$ change of I.E.

$$= 354{\cdot}9 - 69{\cdot}8 + {\cdot}97(759{\cdot}5) - {\cdot}75(972{\cdot}9)$$
$$= \underline{291 \text{ B.Th.U. per lb.}}$$

Alternatively by method (1): Assuming $PV^n = C$, and that n is constant throughout the entire range, then

$$P_1V_1^n = P_2V_2^n$$

or $$n = \frac{\log. P_1/P_2}{\log. V_2/V_1}$$

We have $P_1 = 200 \times 144$ lb.-sq. ft.
$P_2 = 1 \times 144$ lb.-sq. ft.
From tables $V_{s1} = 2{\cdot}29$ cub. ft.
$\therefore V_1 = {\cdot}97 \times 2{\cdot}29$
V_{s2} 333 cub. ft.
$\therefore V_2 = {\cdot}75 \times 333$

Hence $$n = \frac{\log. 200}{\log. 112{\cdot}8} = 1{\cdot}123.$$

$\therefore$ Work done

$$= \frac{144}{778}\left[\frac{(200 \times {\cdot}97 \times 2{\cdot}29) - ({\cdot}75 \times 333)}{1{\cdot}123 - 1}\right]$$
$$= \underline{294 \text{ B.Th.U.}}$$

PROBLEM. Steam at a temperature of 400° F. ($L = 827$ B.Th.U.) expands adiabatically to a temperature of 100° F. ($L = 1{,}036$ B.Th.U.) so that the dryness fraction is the same as at the beginning. Determine this dryness fraction.

Solution. $x = 0{\cdot}476$.

MACHINE CONSTRUCTION AND DRAWING

By G. WOOD, A.M.I.MECH.E.

LESSON XII

Method in Construction. We referred, in the last lesson, to the purpose served by drawing the details of a machine part assembled. We may now proceed to consider the method of producing the finished drawing, combining the principles of machine construction with those of machine drawing. Using the detail drawing, Fig. 3, Lesson XI, as a reference, we may proceed to discuss the method of producing a working drawing of a $2\frac{1}{2}$ in. bore relief valve for a pump discharge.

The pump discharge pressure is limited to 110 lb. per sq. in. and, to suit the arrangement of pump connections, the valve is to be fitted to the branch piece shown in Fig. 1.

It will be observed that the $2\frac{1}{2}$ in. bore branch lies in the horizontal plane and, for simplicity of parts, we would arrange the centre line of the relief valve in the same plane but for an objection in horizontal alignment. Spring loaded valves when arranged in any position other than the vertical tend to jamb, and, consequently, it is usual to arrange the valve with its centre line vertical.

Commencing with the valve body, we may fix the opening and facing to take the valve seat. Proceeding to the valve seat we have a case where rigidity and reliability against wear rather than strength determine the dimensions. We may, therefore, allow the same thickness of metal and flange width as given on the reference drawing. In determining the dimensions of the valve, it being spring loaded, we may ignore the effect of weight and, to obtain the best proportion of strength and rigidity, follow the reference drawing for thicknesses.

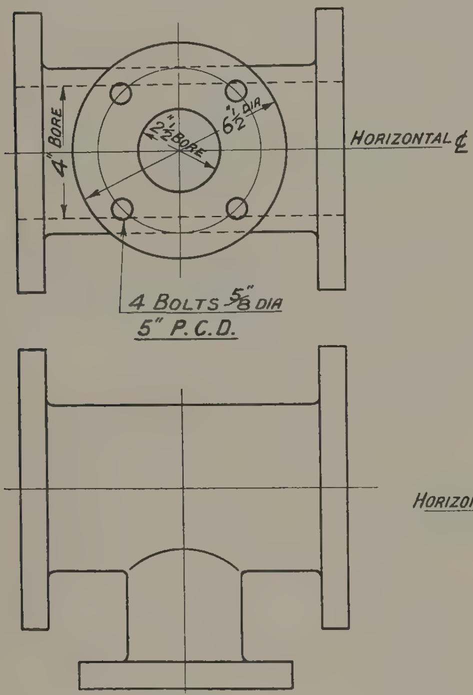


FIG. 1

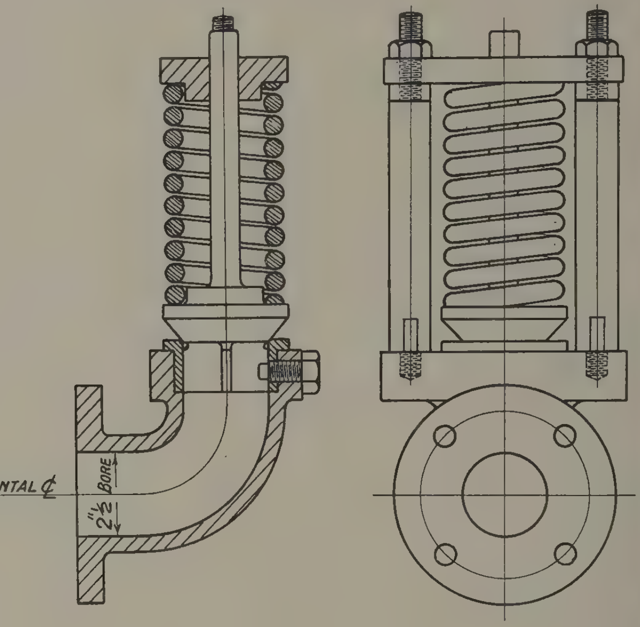


FIG. 2

Adjoining the valve we have the spring, and the dimensions of this part containing more than one variable will require some attention.

Standard Rules. It often happens that empirical rules of proportion have been derived by engineering bodies, such as the various surveys, and although the detail under consideration may not require adherence to any such rule, the similarity of service specified for the detail may induce the designer to use the rules. An opportunity of such appropriation is provided in the design of the above helical spring. The Board of Trade specify that relief valves coming under their survey shall have steel spiral springs designed according to the following rule—

where

D = mean diameter of coil in inches.
d = diameter of section of wire in inches.
s = side of square of wire in inches.
S = load on spring in pounds.
N = number of free coils in spring.
K = compression in inches.
d^1 = diameter of section of wire in sixteenths of an inch.
s^1 = side of square of wire in sixteenths of an inch.

$$d = \sqrt[3]{\frac{S \times D}{8000}} \qquad s = \sqrt[3]{\frac{S \times D}{11{,}000}}$$

$$N = \frac{K \times 22 \times d_1^4}{S \times D^3} \qquad N = \frac{K \times 30 \times s_1^4}{S \times D^3}$$

The governing factor determining the dimensions of the spring is the excess pressure required to give the valve sufficient lift to pass the whole of the discharge which normally flows past the valve. Suppose the pump discharge in the above example is 250 gal. per min., and the allowable excess pressure to be limited to 10 per cent above the normal maximum pressure. We have the maximum release pressure $= 110 \times 1{\cdot}1 = 121$ lb.-sq. in., and the equivalent head (ignoring losses)

$$= \frac{121 \times 144}{62{\cdot}3} = 280 \text{ ft.}$$

the corresponding velocity v

$$= (64{\cdot}4 \times 280)^{\frac{1}{2}} = 134 \text{ ft./sec.}$$

Also, the area of passage between the valve and seat in square feet

$$= \frac{\text{lift} \times \text{circumference of valve}}{144}$$

$$= \frac{2{\cdot}5\pi}{144} \times \text{lift}$$

$$= \frac{\text{quantity passing in cub. ft. per sec.}}{\text{velocity of flow in ft.-sec.}}$$

$$\text{then lift} = \frac{144}{2{\cdot}5\pi} \times \frac{250}{6{\cdot}23 \times 60 \times 134} = (\text{in.})\ {\cdot}0915$$

Tentatively making D, as in the reference drawing, equal to the valve diameter $+ \frac{1}{2}$ in.,

$$\text{we have } d = \sqrt[3]{\frac{3S}{8000}}$$

$$\text{where} \quad S = 110 \times {\cdot}7854 \times 2{\cdot}5^2 = 540 \text{ lb.}$$

$$d = \sqrt[3]{\frac{3 \times 540}{8000}} = (\text{in.})\ {\cdot}58$$

say, (in.) $\frac{9}{16}$

$$\text{also } N = \frac{{\cdot}0915 \times 22 \times 9^4}{540 \times {\cdot}1 \times 3^3} = 9{\cdot}1$$

say, 9 free coils.

Proceeding to the columns, we observe they are always in a state of tension, and that the smallest section resisting the load is at the bottom of the thread in the screwed portion of the part. Analysing the reference drawing where the maximum spring load is 780 lb., we have, where

f_t = maximum tensile stress in lb. per sq. in.
W = maximum load on spring in lb.
A = sectional area of column at bottom of thread in sq. in.

$$f_t = \frac{W}{2A} = \frac{780}{2 \times {\cdot}304} = 1280 \text{ lb. per sq. in.}$$

then for equal strength of column in the $2\frac{1}{2}$ in. valve,

$$A = \frac{594}{2 \times 1280} = {\cdot}232 \text{ sq. in.}$$

say, $\frac{5}{8}$ in. diameter, Whitworth.

Coming to the design of the bridge, we may consider this as being simply supported at the ends and carrying an equally distributed load.

Let l = distance between columns in in.
B = width of bridge in in.
b = diameter of guide hole in in.
d = minimum depth of bridge in in.
f = maximum stress induced in lb. per sq. in.

Then analysing the reference drawing, we have

$$f = \frac{Wl}{8} \times \frac{6}{(B-b)d^2} \text{ approx.}$$

$$f = \frac{780 \times 6{\cdot}25 \times 6}{8 \times 3{\cdot}5 \times {\cdot}75^2} = 1830 \text{ lb. per sq. in.}$$

then, for equal strength, the depth of bridge in the smaller valve will have to be

$$d = \left(\frac{594 \times 5{\cdot}5}{8} \times \frac{6}{(3{\cdot}75 - {\cdot}875) \times 1830}\right)^{\frac{1}{2}}$$

$$d = {\cdot}465^{\frac{1}{2}} \text{ in.} = {\cdot}68 \text{ in., say, } \tfrac{11}{16} \text{ in.}$$

EXERCISE

We leave it as an exercise for the student to proceed with the construction of the $2\frac{1}{2}$ in. relief valve, and make a finished detail drawing similar in style to Fig. 3, Lesson XI.

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XII

TOOLS FOR DRILLING AND BORING

The Drills. Probably no single tool is used so frequently in the machine shop as the drill, of the twist design. The solid twist drills, that is, those with cylindrical shanks, date from 1860, but those made from flat bar, twisted by the smith, were in use long previously, and are not wholly abandoned. The object of the spiral was to favour the escape of chips, but the pitch of the spiral was too short and the chips choked. Another unfavourable feature which it shared with the flat lipped drills, then nearly universal, was that it had no front cutting angle, and therefore only scraped. Both forms were weak, and incapable of rapid rotation and heavy feeding. Twisted drills are made now from flat bar, and are successful, but they are coarsely pitched, and have front rake.

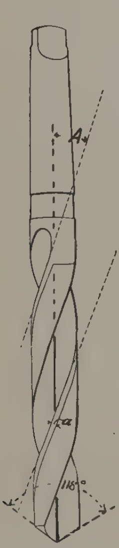


Fig. 71. Twist Drill

Elements of the Twist Drills. Although these are eighty years old, and many thousands of experiments have been made with them, absolute standardization does not exist, except that of the lip angle of 59° with the axis. As the same drills are used for all materials, the copper alloys excepted, the drills for which have straight flutes, the cutting angles are a suitable average. The principal one—the cutting angle—is provided by the helix or twist, Fig. 71, from point to shank. This varies in tools by different manufacturers. An average angle *A* is $27\frac{1}{2}°$ with the axis. The increase twist drills are those in which the angle increases by from 5° to 7° from the point to the shank, with the object of favouring the escape of chips, but the chief advantage is that the strength of the tool is increased near the tang where the torsional stress is greatest. The flutes are milled in solid blanks, previously turned, generally the flutes on opposite sides being done at one time; the blank, held in a chuck, receives a longitudinal and rotary motion past the cutters set at an angle.

The lip angle, 59° with the axis, or 118° included, Fig. 71, is universal, since it yields straight lips, and the tool is more efficient than with smaller angles, which would give convex edges, and with larger angles, that would produce concave edges. The section of the flutes is seen in the end view, Fig. 72, with the point or "chisel edge" at an angle of 135° which is the best for cutting. This is controlled by the angle of clearance given to the lips, generally about 12° at the periphery, Fig. 73, and it should be as high at the centre for easy cutting. As the chisel edge only scrapes, the point is thinned in the larger drills. All the cutting is done with the lips, but the flutes have circular clearance of ½° which does not start from the edge, but a very little way back, leaving "land" *a*, Figs. 71, 72, by which the size is retained. There is a trifle of clearance in diameter from the point to the shank to lessen the longitudinal friction in the

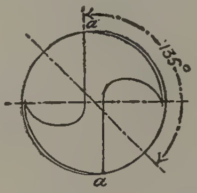


Fig. 72. Section of Flutes of Twist Drill

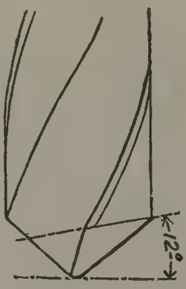


Fig. 73. Lip Clearance of Twist Drill

hole. Drills are held with taper shanks. Morse tapers mostly in drilling machines, sockets being used if the drills are too small to fit the machine spindle. In turret lathes, they fit with parallel shanks.

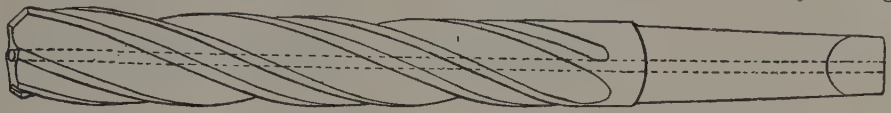


Fig. 74. Four-flute Drill

Drills work most efficiently when they are driven at the highest speed and with the coarsest feed that they will stand. It has been demonstrated that they do their most effective work when the initial sharpness has worn off. Results depend largely on other conditions, so that tables only afford approximations. A driller increases speeds and feeds until he arrives at the best possible for cutting and endurance. Lubrication is the most important condition, subsidiary ones are the support afforded to the work and its mass to carry away the heat, the stiffness of the machine, the fitting of plain or of ball bearings. Drills of plain carbon steel are used less and those of high speed steel which contain large proportions of tungsten and chromium more, and these can be run at from twice to two and a half times the speed of those of straight carbon.

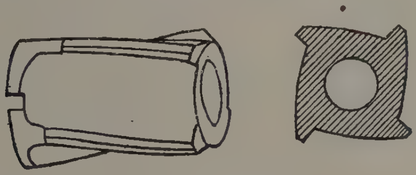


Fig. 75. Shell Drill

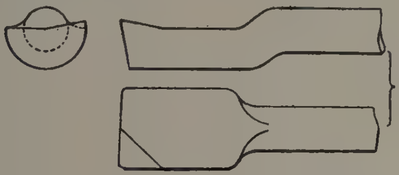


Fig. 76. Solid D-bit

There is an immense variety in the forms of tools which, while they do not initiate holes like the drills, are not boring tools. They enlarge holes, but fill the holes they enlarge. These are the three and four flute drills, Fig. 74, used for enlarging drilled holes and finishing them. Four cutting edges are better than two, and the tool is stiff. One form of this is the shell drill, Fig. 75, short, to fit on an arbor. These are links between the drills and reamers, whose sole functions are the enlargement of holes in very minute degrees. The D-bit is another transitional form that will bore in the solid, or enlarge and form true holes several feet in length, the work being usually rotated and the tool point flooded with lubricant. Fig. 76 is a solid form, Fig. 77 one with an inserted blade. All drilled holes are reamed in accurate manufacture, necessary both when tooling is held to limits, and when holes in adjacent pieces have to be aligned. A reamer never removes much material, say, from $\frac{1}{100}$ in. to $\frac{1}{1000}$ in. These are straight or spirally fluted, of solid or shell forms. For the finest work, the blades are separate from the body and adjustable to permit of retaining the dimensions after wear occurs. Many of these tools are not held rigidly but float in their holders to follow the hole. A selection is given in Figs. 78 to 84.

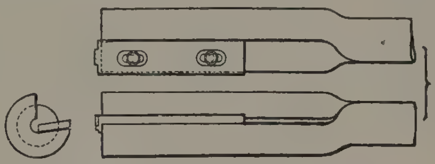


Fig. 77. D-bit with Inserted Blade

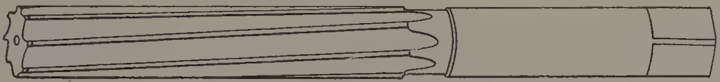


Fig. 78. Solid Reamer Left-hand Flutes

The Boring Tools. The typical boring tool is of round-nose form, solid with its shank, and cranked in the direction of cutting, or a similar tool point fitting and clamped in a slotted bar. When used in the lathe, it is gripped in the tool post and fed by the slide rest. When there is much overhang, say, from 18 to 24 in., even then with a stiff shank or bar the spring of the tool tends to lessen the size of the hole as the farther end is approached. A dull tool will magnify the evil. It is therefore necessary to take a roughing cut first, and measure the bore before finishing. It is better for the smaller holes, say, those up to 6 in. or 8 in. diameter, to use cylindrical bars in the regular machines, not very much smaller than the holes to be bored, carrying one cutter, or several, and driven from one end only, for which from 24 to 30 in. is the useful limit; or supported at both ends for all long bores. The cutters are still round-nose tools for roughing, and straight-faced cutters are used for finishing. They are secured in various ways, Figs. 85–87, with wedges or screws, are single and unbalanced, or double and balanced, and with or without provision for minute radial adjustments. For the larger holes, cutters are disposed in circles, the number increasing with diameters. For the largest work the cutters are fixed in heads mounted on bars, of which there are numerous modifications.

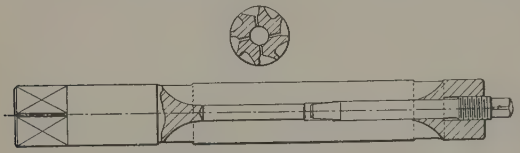


Fig. 79. Reamer Expanded with Tapered Plug Screwed in

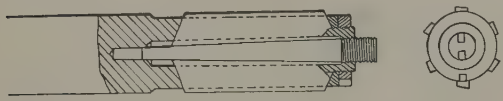


Fig. 80. Reamer with Adjustable Blade

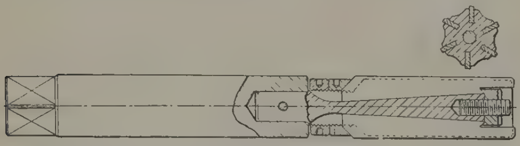


Fig. 81. Reamer Having Blades on Inclined Seatings, Adjusted with Nuts at One End

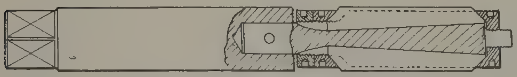


Fig. 82. Reamer Having Blades Adjustable with Tapered Pin and Conical-ended Nuts

When boring is done with a bar carried between the lathe centres, two arrangements are in use. In one, the piece of work is mounted on and clamped down to the carriage of the slide rest, with suitable packings, from which the upper parts have been removed, and the carriage and work are fed longitudinally past the revolving cutters. In the other, the cutters are carried in a head that is splined to the bar, along which it is fed with a screw, operated with a star wheel, or differential gear wheels, the work remaining stationary. This is suitable for the longest pieces and the largest diameter that a lathe will take. But much has been appropriated by the horizontal spindle boring

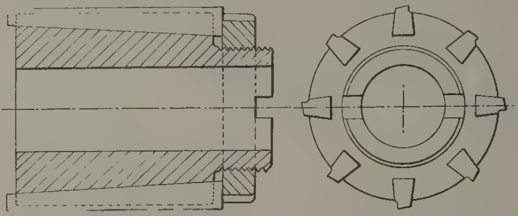


Fig. 83. Reamer Having Blades of Wedge Section in Inclosed Seatings. Adjusted with a Nut

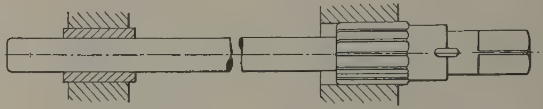


Fig. 84. Shell Reamer, on Pilot Bar

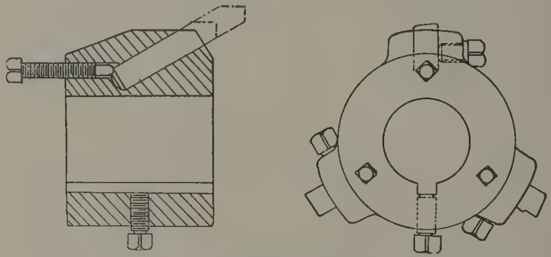


Fig. 85. Adjustment of Cutters in Boring Heads

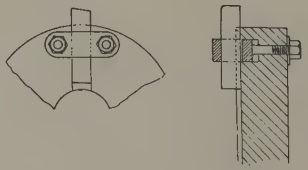


Fig. 86

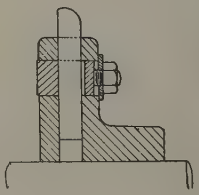


Fig. 87.

Fixing Cutters in Boring Heads

machines, which have the great advantage that the boring heads have provision for making vertical adjustments to take articles of various depths, while in the lathes packing up is necessary.

In the star wheel design of bar, Fig. 88, the feed is effected with a wheel at the end of the screw, that strikes a projection on the bed or poppet at each revolution of the bar, and so turns the feed screw through a small arc. The screw is threaded in a nut within the sliding cutter head, pulling the latter along while it is rotated with the spline. The screw is sunk flush in a recess in the bar. For the largest bars, the feed is effected with spur gears of the epicyclic or the differential type. In the first, a pinion on the poppet mandrel turns a spur gear on the end of the feed screw, the relative sizes of the gears determining the feed. In the second, there are two pairs of gears with a difference of from four to six teeth in the pairs. Two wheels are on the bar, and two on a supplementary shaft. The rate of feed per revolution is constant, but a difference in roughing and finishing cuts is made by varying the speed of revolution of the bar.

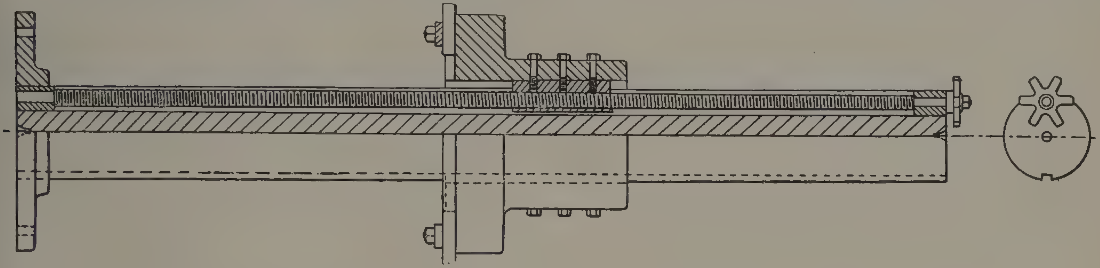


FIG. 88. BORING BAR WITH STAR FEED

Taper boring is either shorter or longer. For very short tapers the slide rest is employed. The tool post is swivelled to the angle and fed by hand. The angle can be set by graduations on the rest if these are available, and if not, then by setting out the angle or the taper per foot and transferring it with a bevel to the rest. After a rough cut the angle should be checked before finishing. To bore tapers with a bar between centres, the essential is, that the bar must rotate on an axis coincidently with the angle of the bore. It is not set over with the loose poppet centre, since that would produce a parallel hole. It is set over radially from the centre to the angle wanted. A special fitting is required to hold the bar, its radial screw and its feed gears, Fig. 89. It is centred on a headstock centre bolted to the face-plate and driven with dogs. The other end runs on the centre of the loose poppet. Being set at an angle, a spherical block in the bar receives the centre point, and accommodates itself to angles. Large holes are better bored tapered on the vertical mills if they come within the capacity of the machine.

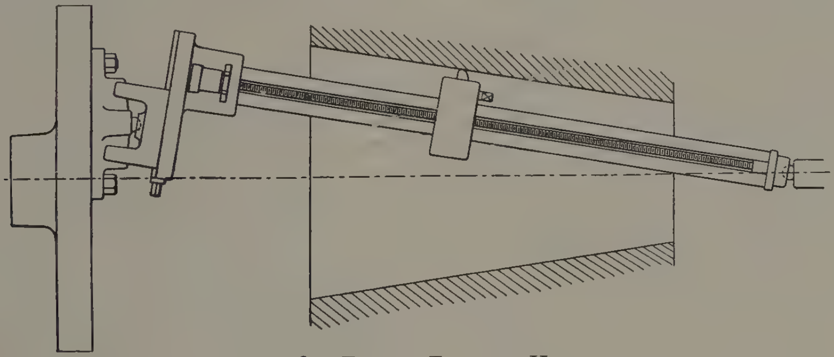


FIG. 89. BORING TAPERED HOLES

DIESEL ENGINES
CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON VII

MECHANICAL DETAILS OF THE FOUR-STROKE AIR INJECTION ENGINE

In dealing with this part of the subject we are compelled, by reasons of space, to describe in detail only those individual parts which are peculiar to the Diesel engine. Parts which are common to all engines of similar mechanical form must be passed over with as little comment as possible. For the same reason the special parts that will be described are those belonging to the most commonly accepted design of engine, and are chosen mainly for the purpose of illustrating the essential features. In practice, of course, considerable variation in design exists as between the engines of the different makers.

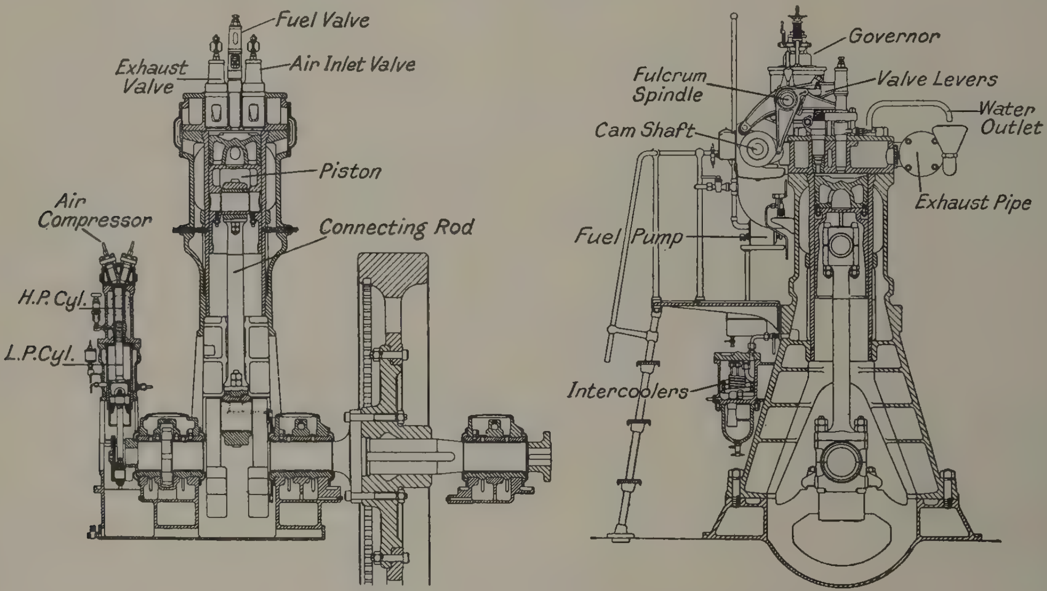


Fig. 12. Single-cylinder, Trunk-piston Type Engine (Mirrlees), 50 H.P.

We give first, in Fig. 12, a sectional drawing of a single-cylinder, open-frame, trunk piston type engine which may be regarded as the standard form for small and medium sizes and, in fact, is the prototype of all commercial Diesel engines. The main variations from this type are engines with completely enclosed working parts, and engines with crossheads, the latter being either of the open or the enclosed type and more especially used for larger sizes. From this drawing and from the photograph of an actual engine shown in Fig. 13, the general disposition of the principal parts and their relationship to the engine as a whole can readily be seen.

Cylinder Cover, Valves, and Valve Gear. The cylinder cover is a plain circular casting of box section (see Fig. 14), in which are provided four bored holes with seatings for the air (*A*), exhaust (*B*), fuel (*C*), and starting (*D*) valves and passages at *E* and *F* for the air and the exhaust gases respectively. The cover is water cooled, and is held to the cylinder frame *H* by a number of studs or bolts, *G*, which take the direct load from the gas pressure in the cylinder and serve to keep gas-tight the spigotted joint *J* between the cover and the cylinder. This joint is usually made with a narrow flat ring of copper and asbestos. The material of the cylinder covers is almost invariably cast iron, though cast steel has been used to some extent. The general construction is made, perhaps, clearer in Fig. 15, showing an actual cover and its valves in section.

Fig. 16 illustrates the exhaust valve *A*, complete with its operating lever *B* and cam *C*. The valve itself is made of nickel steel, while the casing is of cast iron and in two parts, the lower half or seating *D*, and the upper half or cover *E*, which acts as a guide for the top of the valve spindle.

The lever *B* is either of cast iron or steel, and is so constructed that the end over the valve can be detached or swung back on a hinge in order to facilitate the removal of the valve. At the valve end there is a swivel pin *G*, fitted with the adjustable tappet screw *H*, while at the cam end the lever carries a hardened steel roller *J*.

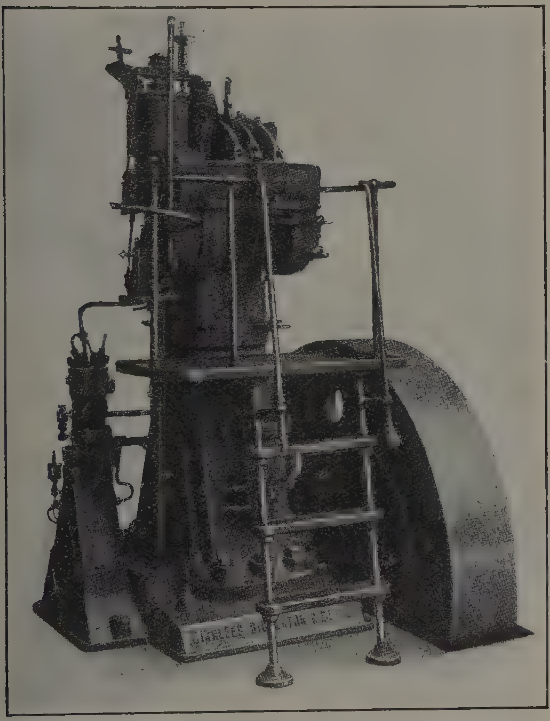


FIG. 13. SINGLE-CYLINDER 50 B.H.P. "MIRRLEES-DIESEL" ENGINE

In large engines positive cooling of the exhaust valve parts becomes necessary, and Figs. 17 and 18 show two alternative designs. In the first the casing only is cooled, while in the second the actual valve spindle has water flowing through it, the water being carried to and from the top end of the spindle by means of flexible pipes. The cast-iron seat on the spindle in Fig. 17 should also be noticed as it is now common practice to fit the valve with a cast-iron head fixed by mechanical means as shown, or either cast or welded solid with the valve spindle itself. Cast iron has proved itself a very suitable material for withstanding the conditions under which an exhaust valve has to work.

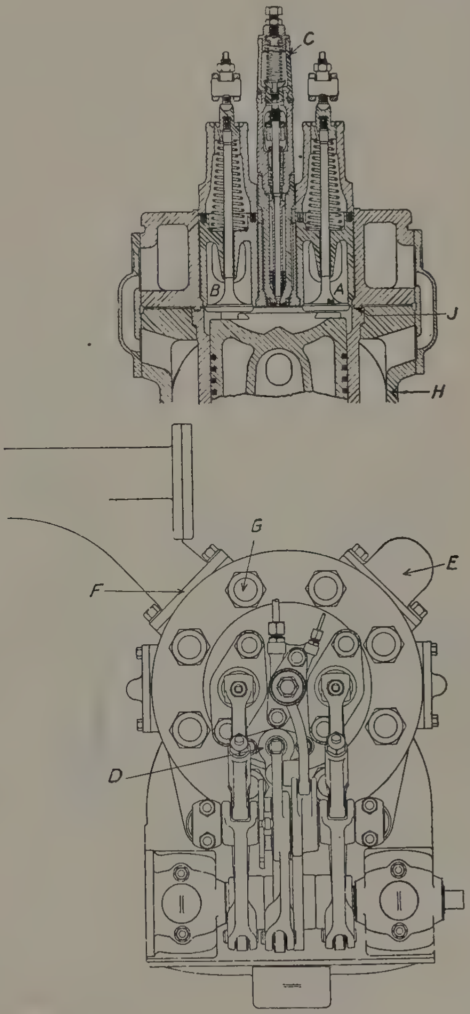


FIG. 14. CYLINDER COVER WITH VALVES

The air valve with its operating gear is of similar general design to the exhaust valve, except that even on very large engines no water cooling is necessary, the incoming fresh air effectively keeping the parts at a workable temperature.

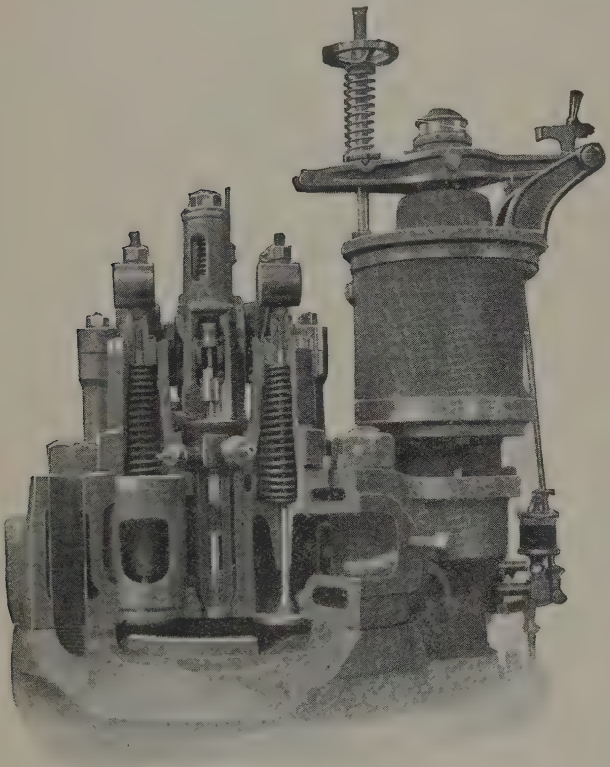


FIG. 15. SECTION THROUGH CYLINDER COVER AND VALVES

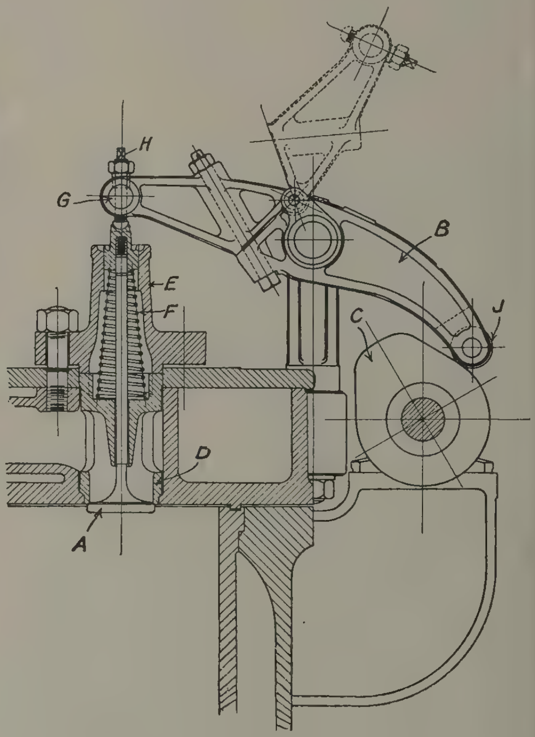


FIG. 16. EXHAUST VALVE, LEVER, AND CAM

The fuel valve and starting valve, together with their operating levers and cams, are shown in Fig. 19, while Figs. 20 and 21 give the details of the valves themselves to a larger scale. The fuel valve has a bottom casing *A*, Fig. 20, seated in the cylinder cover, bored out to receive the needle valve *B*, and the pulverizer sheath *C*, and drilled to admit the fuel oil at *D*, and the blast air at *E*. A middle casing *F* fits into the bottom casing and provides a guide *G*, and stuffing box *H*, for the needle valve itself, while a flange cast on it serves as a means of securing both casings to the cover by studs. The top casing *J* houses the spring *K*. The pulverizer sheath or tube *C*, fitting around the valve spindle and inside the bottom casing, carries at its lower end a series of drilled and grooved washers *P*, and a conical, grooved nut, and screwed on to the sheath and holding the washers in position. The flame plate or nozzle *R* is held in position by a nut *S* screwed on to the threaded extension of the bottom casing.

The operation of the fuel valve is as follows: The annular space *T* is kept full of high pressure blast air, which is supplied through the pipe *Y*. The fuel oil is pumped in through the small hole *U* just above the pulverizer washer, the correct quantity being delivered once in each cycle. On entering, the oil lies more or less evenly around the top of the upper washers. When the needle valve is lifted the blast air rushes into the cylinder carrying the oil with it, first through the small holes in the washers, second through the grooves in the nut, then through the narrow annular and tapering space between the needle valve point and the conical seating, and finally out through the nozzle into the cylinder. The function of the so-called pulverizer washer and nut is really to mix the air and the oil thoroughly together by providing a long and tortuous path and a large surface on which the oil tends to cling. The blast air passing with high velocity over these surfaces tears off the clinging oil, breaking it up thoroughly and distributing it more or less evenly amongst the air. The final pulverization takes place at the nozzle itself, the correct shape and size of which is the most vital factor of successful pulverization. The combined resistance of the washers, nut, valve opening, and nozzle, regulates the rate at which the oil and air enter the cylinder, and it is by the manipulation of these factors that the correct rate is found to give approximately constant pressure combustion with the large variety of oils in practical use. Experiment and experience only can determine these values.

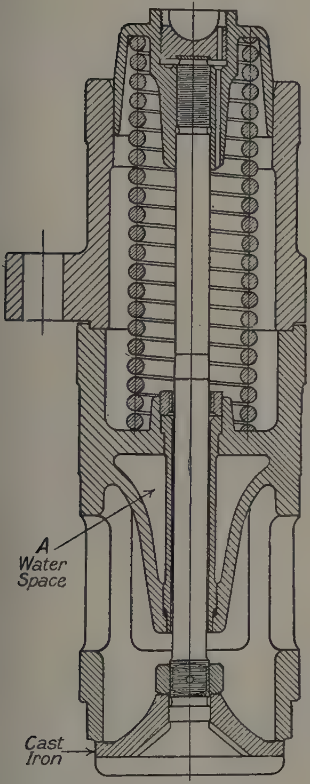


Fig. 17. Exhaust Valve with Water-cooled Casing and Cast-iron Face

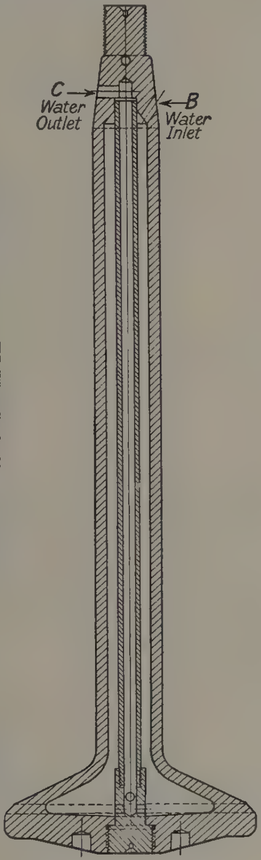


Fig. 18. Exhaust Valve with Water-cooled Spindle

The lift of the fuel valve is quite small, varying from about $\frac{1}{16}$ in. on an engine of, say, 7 in. diameter cylinder, to about $\frac{1}{4}$ in. or $\frac{5}{16}$ in. with a cylinder of, say, 30 in. diameter.

The starting valve is very simple. Its construction is made clear by Figs. 19 and 20, from which it is seen that the valve opens inwards and is returned to its seat by a spring, which has to be strong enough to do this against the starting air pressure. It is not customary to fit packing around the spindle as the long guide surface is sufficiently tight to prevent any considerable leakage of air during the short starting period. The casings are of cast iron, while the spindle is sometimes of bronze and sometimes of steel with a white metal sleeve.

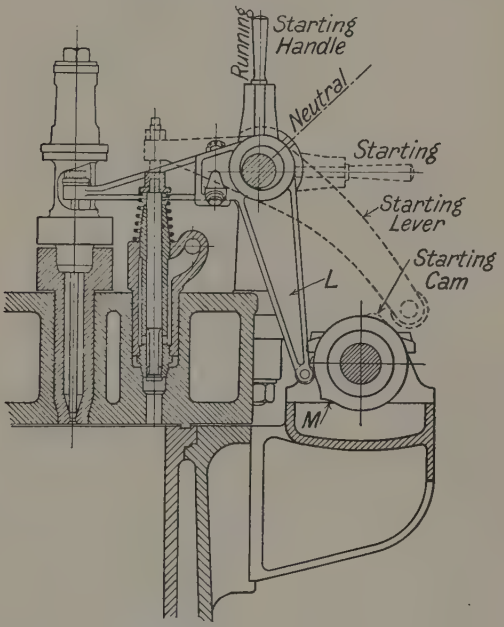


Fig. 19. Fuel and Starting Valves with Levers

Starting Gear. The arrangement by which the fuel and starting levers are alternately thrown out of contact with their cams is shown generally in Fig. 19 and diagrammatically in Fig. 22. On the fulcrum spindle *A*, Fig. 22, which is a plain shaft supported on brackets from the cylinder cover or other fixed part, there is mounted an eccentric sleeve *S*, upon which the fuel and starting levers are carried. To this sleeve is attached a handle or lever which can be fixed by means of a quadrant in any of the three positions corresponding to "running," "neutral," and "starting" respectively. These positions are shown at *D*, *C*, and *B*, from which it is seen that in the "running" position only the fuel lever is in contact with its cam; in the "neutral" position neither lever is engaged; and in the "starting" position the starting lever has come into action, and the fuel lever is inoperative.

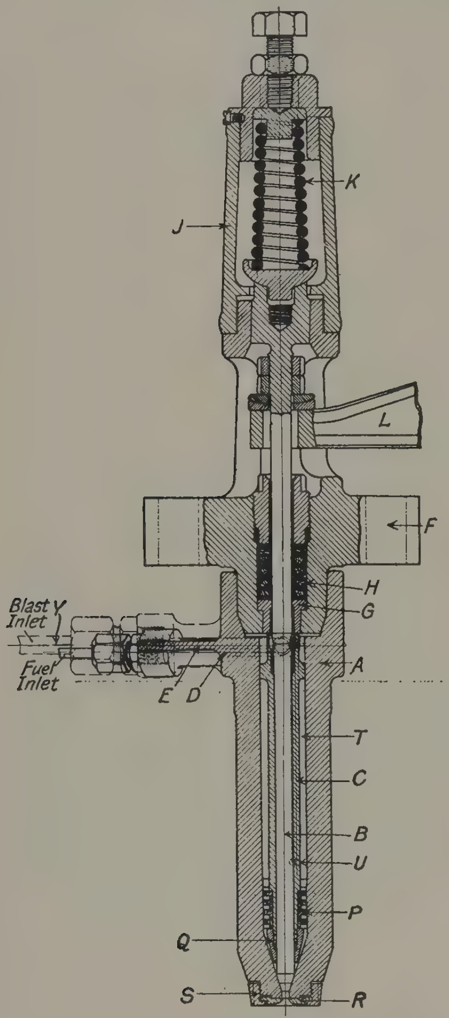


FIG. 20. DETAIL OF FUEL VALVE CASING AND NEEDLE VALVE

The above is the usual arrangement for individual cylinders. In multi-cylinder engines, where one or more of the cylinders may be fitted with starting valves, the fulcrum spindles of all the cylinders so fitted can be joined together and operated by one lever, in which case the eccentric portion is fixed to the shaft and moves with it. The exhaust and air levers are mounted on the concentric part of the fulcrum spindle.

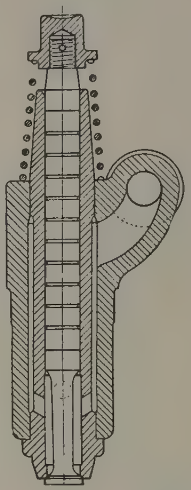


FIG. 21. DETAIL OF STARTING VALVE

Cams and Camshaft. The cams are designed to give the correct opening of the valves in regard to the timing and lift, together with smoothness of operation. A typical set of cams gives the following timings—

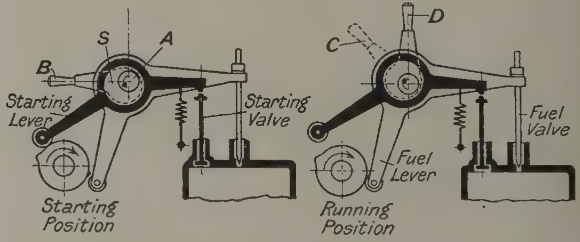


FIG. 22. DIAGRAMMATIC REPRESENTATION OF STARTING GEAR

Valve	Opens	Closes
Fuel	5° before top centre	25° after top centre
Exhaust	40° before bottom centre	12° after top centre
Air	10° before top centre	10° after bottom centre
Starting	At top centre	70° after top centre

The cams are usually made of cast iron, though sometimes of steel. In the case of the fuel cam the small sector forming the actual profile is made in a separate piece of hardened steel, and provision is made for slight angular adjustment, as indicated in Fig. 23 showing a common form of this construction. The cams are keyed to a shaft running along the front of the cylinders in the position shown clearly on the sectional drawings and photographs already given. In some designs, however, the shaft is much lower, and the motion is transmitted to the valve levers by means of push rods. In the former case the camshaft is driven from the crankshaft by skew gear wheels and a vertical intermediate shaft, while in the latter a direct spur gear drive can be and sometimes is adopted.

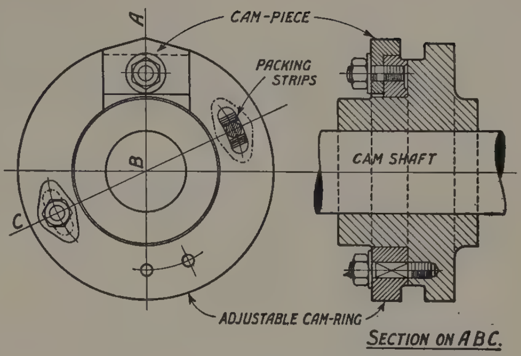


FIG. 23. FUEL CAM SHOWING METHOD OF ADJUSTMENT

STEAM TURBINES

By W. J. KEARTON, M.ENG., A.M.I.MECH.E., A.M.INST.N.A.

LESSON II

CLASSIFICATION OF STEAM TURBINE TYPES

ALTHOUGH the fundamental principle on which all steam turbines operate is the same, yet the methods whereby this principle is carried into effect vary, and, as a result, certain types of turbines have come into existence. The precise differences between the existing types is often imperfectly understood, and it is hoped that the following brief classification will be of some value.

The Simple Impulse Turbine. If the two words *simple* and *impulse* are clearly understood, then, if the name is a logical one, it ought to describe exactly on what principle the turbine is constructed.

Simple means that the steam is expanded *once* only between the initial and exhaust pressures.

Impulse indicates that the steam thrust on the blade is produced by a change of velocity (which may be brought about by a change of direction and not necessarily by a change of magnitude) *at constant pressure.* The pressure of the steam may, and does actually, vary within the blade channels, but the pressure at outlet is sensibly the same as that at inlet. In other words, there is no expansion of the steam, as a whole, within the moving blades.

Fig. 3 shows diagrammatically the type under consideration. In its more usual form, it comprises a single wheel or disc mounted on a spindle and supporting the blades at its periphery. The variation of pressure and velocity are shown in the lower part of the diagram. It will be seen that the pressure of the steam falls from boiler to condenser pressure entirely in the nozzles. The velocity reaches its maximum value at exit from the nozzles and is reduced during the passage of the steam through the blades.

Efficiency of Simple Impulse Turbine. Consider the two velocity diagrams in Fig. 4. In (*a*), the blade velocity u has been made small compared with the initial steam velocity V_1, with the result that the change of velocity AF and, therefore, the useful component AG, which is usually called the *velocity of whirl*

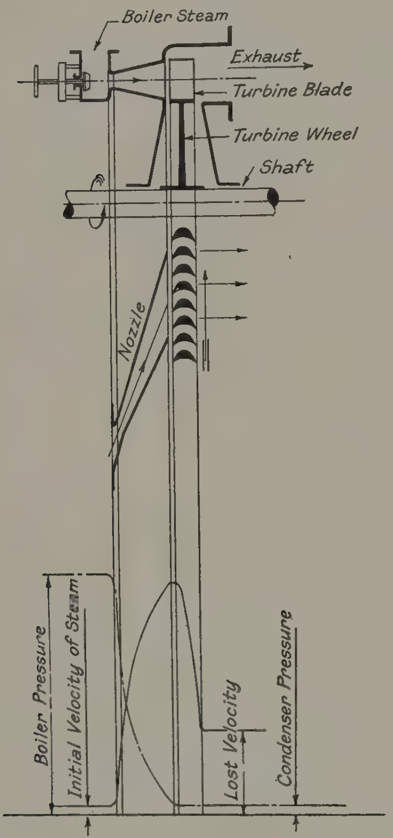


FIG. 3. DIAGRAMMATIC ARRANGEMENT OF SIMPLE IMPULSE TURBINE

V_w, is relatively large. But the thrust on the blades is equal to

$$\frac{w \, . \, V_w}{g} \text{ lb. weight}$$

where w = weight of steam flowing per second.
,, V_w = velocity of whirl in ft. per second.

Now the work done = (thrust) × (blade velocity) and here we have a large thrust multiplied by a small blade velocity.

In Fig. 4(*b*), the blade velocity u has been made relatively large and the result is a comparatively small velocity of whirl and a small thrust.

If the blade velocity u be varied while keeping the steam velocity V_1, and the nozzle angle α constant, it will be found that the product

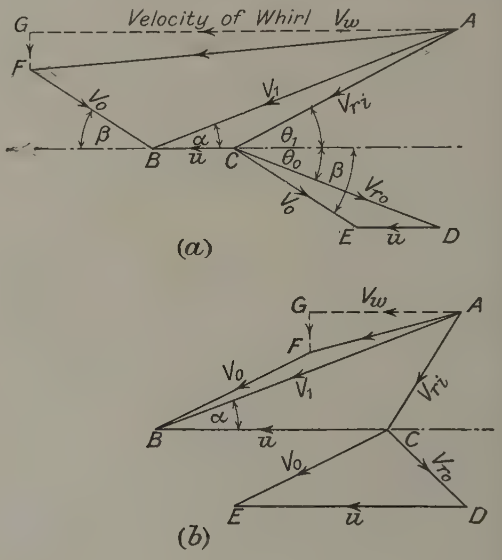
(thrust) × (blade velocity),

which is equal to the useful work done by w lb. of steam, will vary considerably, and there is a certain value of the ratio $\frac{u}{V_1}$ for which the work done becomes a maximum.

Referring to Fig. 4 (*a*), it will be clear that

$$\begin{aligned} V_w &= V_1 \, . \cos \alpha + V_o \, . \cos \beta \\ &= V_1 \, . \cos \alpha - u + V \ \ . \cos \beta + u \\ &= V_{ri} \, . \cos \theta_1 + V_{ro} . \cos \theta_o \\ &= V_{ri} \, . \cos \theta_1 \left\{ 1 + \frac{V_{ro} \, . \cos \theta_o}{V_{ri} \, . \cos \theta_1} \right\} \end{aligned}$$

The value of the fraction $\frac{V_{ro} \, . \cos \theta_o}{V_{ri} \, . \cos \theta_1}$ depends upon the ratios $\frac{V_{ro}}{V_{ri}}$ and $\frac{\cos \theta_o}{\cos \theta_1}$. The former depends upon the losses caused by surface friction and internal friction in the steam. It is, of course, always less than unity. We shall represent it by the symbol k, and we shall further assume that k is sensibly constant. In actual turbines the outlet angle θ_o is generally a few degrees smaller than θ_1, but the ratio of the cosines may be assumed to be constant



(*a*)

(*b*)

FIG. 4

without any appreciable degree of error. Hence, we may write—

$$C = \left\{1 + \frac{V_{ro} . \cos \theta_o}{V_{ri} . \cos \theta_1}\right\} = \left\{1 + \frac{k . \cos \theta_o}{\cos \theta_1}\right\}$$

and assume C to be a constant.

$$\begin{aligned} \therefore \quad V_w &= C . V_{ri} . \cos \theta_1 \\ &= C(V_1 . \cos \alpha - u). \end{aligned}$$

But the work done *per lb. of steam*

$$= \frac{V_w . u}{g} \text{ ft.-lb.}$$

and since the initial kinetic energy of the steam (per lb.) is $\frac{V_1^2}{2g}$, then the efficiency η must be equal to

$$\eta = \frac{\text{work done}}{\text{initial kinetic energy}}$$

$$= \frac{V_w . u}{g} \times \frac{2g}{V_1^2} = \frac{2 . V_w . u}{V_1^2}$$

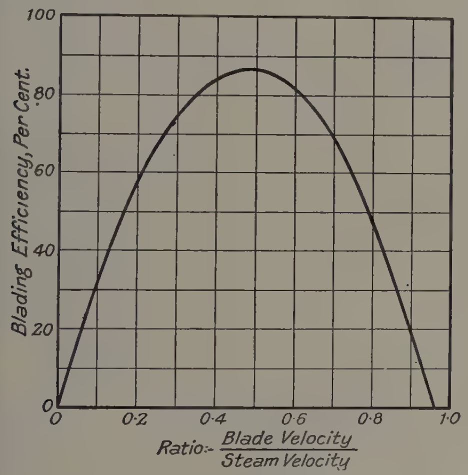


FIG. 5. BLADE EFFICIENCY CURVE FOR IMPULSE TURBINE
(Equiangular blades)

Substituting the value of V_w in this equation, we have

$$\eta = \frac{2C(V_1 . \cos \alpha - u) . u}{V_1^2}$$

The most important ratio in all steam turbine calculations is that of blade velocity to steam velocity. We shall represent this ratio by the symbol ρ (the Greek letter *Rho*, pronounced rō).

i.e. $\rho = \frac{u}{V_1}$, or $u = \rho . V_1$.

$$\therefore \quad \eta = \frac{2 . C . V_1^2 . (\rho . \cos \alpha - \rho^2)}{V_1^2}$$

$$= 2 . C\ (\rho . \cos \alpha - \rho^2).$$

Values of the efficiency η have been plotted in Fig. 5, for $\alpha = 16°$, $\theta_1 = \theta_o$, and $k = 0·88$.

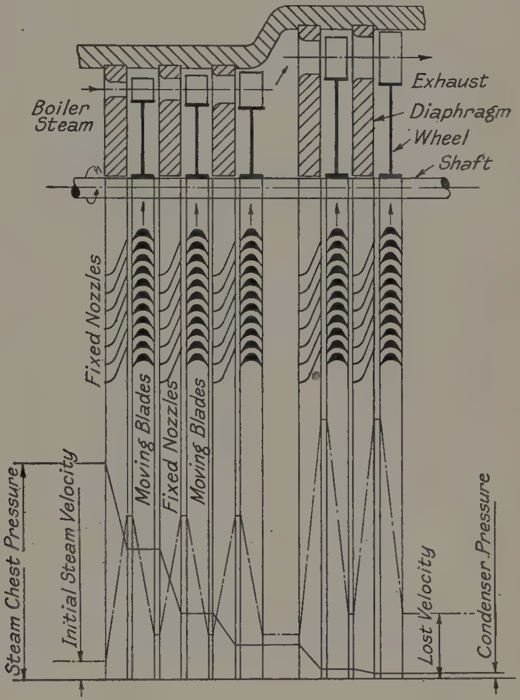


FIG. 6. DIAGRAMMATIC ARRANGEMENT OF PRESSURE-COMPOUNDED IMPULSE TURBINE

It will be observed that the efficiency reaches a maximum value when ρ is about 0·48. The exact value may be found as follows—

$$\frac{d\eta}{d\rho} = \cos \alpha - 2\rho.$$

For maximum efficiency, $\frac{d\eta}{d\rho} = 0$.

$$\therefore \cos \alpha - 2\rho = 0, \text{ or } \rho = \frac{\cos \alpha}{2}.$$

Thus, if the nozzle angle is 16°, $\rho = 0·4806$; if $\alpha = 20°$, $\rho = 0·4698$; if $\alpha = 30°$, $\rho = 0·433$ for maximum efficiency.

Pressure-compounded Impulse Turbine. If the heat energy available by the adiabatic expansion of the steam in a turbine happens to be fairly large, the blade velocity will be very high. For instance, dry and saturated steam expanding from a pressure of 215 lb. per sq. in. absolute to a pressure of 0·5 lb. absolute will liberate 371 B.Th.U.'s. Assuming no losses, the velocity of the issuing jet of steam would be 4,310 ft. per second, and the theoretically most efficient blade velocity would be over 2,000 ft. per second. Apart from the fact that it is impossible to construct a wheel of sufficient mechanical strength for this high peripheral velocity, it would involve either a very large disc or a very high speed of rotation.

This difficulty can be avoided by causing the total drop in pressure to take place in a number of stages, i.e. *by pressure-compounding*. The pressure-compounded turbine, which is shown diagrammatically in Fig. 6, is simply a number of simple impulse turbines on a common spindle and working in series.

EXAMPLE. The mean diameter of the blades of an impulse turbine is 45 in., the speed of rotation is 3,000 rev. per minute, the nozzle angle is 20°, and the velocity of the steam at exit from the nozzles is 1,440 ft. per second. Draw the diagram of velocities, assuming that the relative velocity at exit from the blades is 0·85 times the relative velocity entering the blades, and that the blade angles are equal. Obtain or calculate the following—

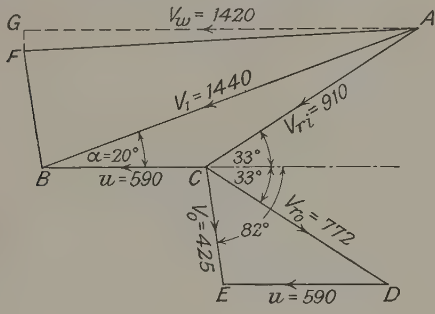


FIG. 7

1. The inlet angle θ_1.
2. The velocity of whirl.
3. The thrust on the blades.
4. The work done per second.
5. The horse-power developed.
6. The efficiency.
7. The axial thrust on the blades.

The weight of steam per second is 5 lb.

$$\text{Blade velocity } u = \frac{\pi \times 45 \times 3000}{12 \times 60}$$

$$= 590 \text{ ft. per second.}$$

The velocity diagram may then be partly drawn, giving $V_{ri} = 910$ ft. per second, and the inlet angle $\theta_1 = 33°$.

Since $k = 0·85$, $V_{ro} = 0·85 \times 910 = 772$ ft.-sec. Completing the diagram, we find that the final absolute velocity of the steam is 425 ft. per second. Drawing *FB* equal and parallel to *CE*, we obtain *FA*, the change of velocity of the steam. This may then be resolved into its components *GA* and *GF* respectively, tangential and axial.

(2) $V_w = 1420$ ft. per second.

(3) Tangential thrust on blades $= \dfrac{w \,.\, V_w}{g}$

$$= \frac{5 \times 1420}{32·2} = 221 \text{ lb. weight.}$$

(4) Work done per second $= 221 \times 590$
$= 130,400$ ft.-lb.

(5) Horse-power developed $= \dfrac{130,400}{550} = 237$.

(6) Efficiency

$$= \frac{\text{work done}}{\text{initial kinetic energy of steam}}$$

$$= \frac{\text{work done} \times 2g}{w \,.\, V_1^2}$$

$$= \frac{130,400 \times 64·4}{5 \times 1440 \times 1440} \times 100 \text{ per cent.}$$

$$= 81 \text{ per cent.}$$

(7) Since $GF = 80$ ft. per second,

$$\text{Axial thrust} = \frac{5 \times 80}{32·2} = 12·4 \text{ lb. weight.}$$

METALLURGY FOR ENGINEERS

By James Knowles

LESSON I

INTRODUCTION

Metallurgy is generally defined as "the art of extracting metals from their ores and of adapting them to the various purposes of manufacture." This presupposes, rather than suggests, the study of the properties of these metals, either in the free state or in association with other metallic or non-metallic elements.

Practised as an art from prehistoric times and as an industry since before the dawn of the Christian era, it is scarcely an exaggeration to state that metallurgy received its greatest impetus as a science from the epoch making discovery of the Bessemer process of steel manufacture in 1856. It is, however, in its industrial application that this science will be treated in these articles.

Even then, the necessary condensation of so wide a field to fit the obvious restrictions of space will permit of no more than a brief survey of the operations concerned and the underlying theories. From time to time standard works of reference will be indicated, in which the engineer, whose interest is sufficiently aroused, may find an exhaustive consideration of the subject, and thus materially increase his knowledge of a science so closely bound up with his own.

Of all the metals placed by Nature at man's disposal none has so contributed to his advancement as iron, and the resultant importance of the iron and steel industry has so focused scientific interest upon and around it that it is usual to consider metallurgy as consisting of two great branches—ferrous and non-ferrous—with the former occupying the more prominent place.

The products of the iron and steel industry may be summarized as follows—

1. Wrought iron,
2. Cast iron,
3. Malleable cast iron,
4. Steel,

the basis of all these being the metal iron.

CAST IRON

Its Properties and the Influence of Its Chemical Constituents Thereon

To the metallurgist the term *cast iron* embraces the metallic product of the blast furnace as well as that of the foundry, but in works practice a distinction is drawn between the two. Iron from the blast furnace is referred to as *direct metal* if received for a subsequent operation in a molten condition, and *pig-iron* if supplied cast in that familiar form, whilst *cast iron* is specially reserved to describe iron made into castings in the foundry. To avoid ambiguity, it is proposed to adopt this system of nomenclature in this series of lessons.

Pig-iron generally consists of about 90 to 95 per cent of iron with which are associated, in varying proportions, "impurities" comprising chiefly carbon, silicon, manganese, sulphur, and phosphorus. All these elements have their individual and characteristic effects, not only on the iron but also upon each other. Therefore, given a knowledge of these effects and an ability to limit and utilize them, it would appear that some of the so-called impurities may almost come to be regarded as essentials.

Carbon is the most important of these constituents and is present, usually to the extent of some 3 to 4 per cent, in two distinct forms: *combined carbon*, which is intensely hard and brittle, and *graphite*, which is soft and tough.

It is the proportion of these two types of carbon which determines the classification of an iron.

Combined carbon forms with the iron a definite chemical compound (Fe_3C), and it is the presence of this carbide which has the predominating influence on the hardness of cast iron. As its percentage rises, in a series of irons, so also do the strength, hardness, and, finally, the brittleness increase. To reduce this to figures, easily memorized if not scientifically accurate to the second decimal, when the percentage of combined carbon does not exceed 0·25 the metal is soft; tensile strength increases to its peak with successive increments up to 0·50; transverse tests are best near 0·75 per cent, and good compression results from 1·0 per cent. Hereabouts, however, the hardness of the metal begins to operate against economical machining and, therefore, castings of grey iron should not contain more than this amount.

Free, or graphitic, carbon exists, mechanically mixed with the iron, as flakes of graphite.

These flakes have a considerable weakening effect on the tensile strength of the metal as, by breaking the continuity of the mass, they set up a series of planes of weakness which yield when the iron is stretched. Conversely, if a compressive force be applied they are simply pressed closer together and so assist resistance to the force. Grey iron is therefore weak in tensile strength, strong in compression and soft to the machine tool or drill.

In so far as all the carbon not in the combined state is graphitic, it can be said that free carbon exercises a softening effect on the metal, but sight must not be lost of the fact, previously stated that it is the combined carbon which is directly responsible for the hardness.

With cast iron in a state of complete fluidity all the carbon present is in the combined state, but during the cooling (solidification), decomposition of the carbide occurs to a greater or less degree. Hence, any factors which tend to advance or retard this decomposition are of vital importance to the ironfounder. What, then, are these factors?

The first is physical—the rate of cooling. The iron carbide is simply a solution of carbon in iron and, as such, obeys the same law as do all solutions. Slow cooling favours the formation of large crystals (graphite), whilst with quick cooling a *solid* solution (combined carbon) can be obtained.

The second factor is chemical and depends on the influence of the several constituents. Of these it is the element silicon which exerts the greatest influence on the decomposition of the carbide.

Silicon. Up to about 2 per cent, silicon has practically no direct effect on the iron itself, but exercises an important effect on the carbon since it tends to keep it in the free, or graphitic, state. The whole physical properties of cast iron depend on the amount and, to an even greater degree, the condition of the carbon present, therefore, the importance of the influence of silicon cannot be over-emphasized. The attention of the student is thus drawn to the results of an exhaustive research on this subject carried out, in 1885, by Professor Turner, details of which are published in his *Metallurgy of Iron.*

The size of the castings, which in a large measure fixes the rate of cooling, and the proportions of the other elements present in the iron have an important effect, however, and, bearing this in mind, for practical working, the undermentioned table of silicon standards, used and advocated in many lectures and publications by Dr. Percy Longmuir, is recommended as a guide—

Type of Casting	*Silicon* Per cent
Malleable cast iron, normal section	0·4
Malleable cast iron, thin section	1·0
Chilled grey iron castings	0·7–1·0
High pressure cylinders, valve bodies, etc.	1·3
Machine and engine details, gearing, etc.	1·5
Soft engineering and millwright castings	2·5
Soft thin castings, stove, grate, and similar work	2·5–3·0
Hollow ware	3·0–3·5

Manganese. The presence of manganese has the effect of replacing, to some extent, the carbide of iron already referred to by a double carbide of iron and manganese, which is closer in texture and stronger than the normal carbide. It thus enables more of the iron to combine with the carbon and tends, in this way, to keep more of the carbon in the combined form. Hence, manganese is regarded as a hardener as distinct from silicon with its softening tendencies.

Sulphur. Sulphur is always regarded as an unmitigated evil. Its presence makes for hardness, brittleness, and unsoundness, since it tends to form sulphide of iron. This, during the cooling of the molten metal, is thrown to the edges of the iron crystals and thus weakens the cohesion between them. Castings should, therefore, be kept as free as possible from this element, with a definite maximum fixed at 0·2 per cent. Manganese and sulphur will, however, react upon each other to form sulphides of manganese, and, as the presence of an excess of manganese favours the formation of manganous sulphide which separates out from the fluid metal, the influence of sulphur can, in this way, be kept in subjection.

Phosphorus. Phosphorus exists in pig-iron as phosphide of iron (Fe_3P), and has no effect on the condition of the carbon. It increases the fluidity of the molten metal and makes it capable of retaining the most intricate features of the mould. For this reason phosphorus contents of as much as 2·0 per cent are permissible for ornamental castings. Phosphorus, however, increases the brittleness of cast iron, weakens it mechanically, and, on account of the slow solidification of the phosphide, makes for unsoundness in the castings and a segregation of phosphorus in the thicker portions.

Summary. The foregoing considerations may then be briefly summarized as follows—other things being equal and within commercial limits—

1. The percentage of combined carbon increases the hardness and brittleness of cast iron as it is itself increased.
2. The carrying power of iron for combined carbon is assisted by the presence of manganese, which effect is, however, neutralized by silicon.
3. The quicker the cooling, the higher the percentage of combined carbon. The higher the silicon, the lower the percentage of combined carbon. Therefore, the quicker the rate of cooling, the more silicon is necessary to limit the percentage of combined carbon to the desired amount.
4. Sulphur, having a greater affinity for manganese than for iron, will, in the presence of a sufficiency of the former, form a sulphide of manganese which is not so harmful in its effects as the iron sulphide.

Finally, it should be noted that at the very moment of solidification cast iron suffers a considerable expansion. This enables it to take unto itself the internal shape of the mould, but once solid, a contraction takes place and continues until the metal is cold. This contraction, or shrinkage, must be allowed for, of course, and as a general rule it can be understood that shrinkage, more or less proportionately, follows the hardness of the cast iron.

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON VII

DRY SAND MOULDING

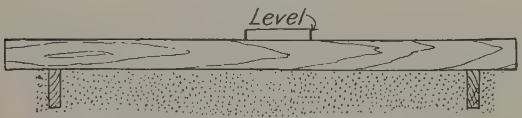
Sand, when used for a mould to be dried, frequently contains more moisture than sand used for green sand moulding, because it facilitates working. The increased moisture makes the sand more plastic, but does not adversely affect the mould, as it is evaporated and the mould made dry before receiving the metal. The majority of medium and large sized castings are made in dry sand moulds, because of the greater security it offers for the production of sound castings. Drying a mould, however, before it receives the fluid metal, necessarily increases its cost and this naturally influences the cost of castings.

Dry Sand Mixtures. Sand to be dried requires more bonded strength than green sand mixtures, in order to make up for the absence of moisture when dried. Thus, whatever kind of metal is cast, the sand for moulds to be dried is stronger than would be used for similar metal in green sand. For steel castings, Yorkshire, Cornish, and Belgian sands are used because of their freedom from impurities. The mixtures used for facing varies with the bonded strength of the sand and to suit the kind of work to be done. China clay may be used to increase the strength. The moulds for large steel castings are generally faced with a special mixture known as "compo," it being less likely to burn on to the surface of castings.

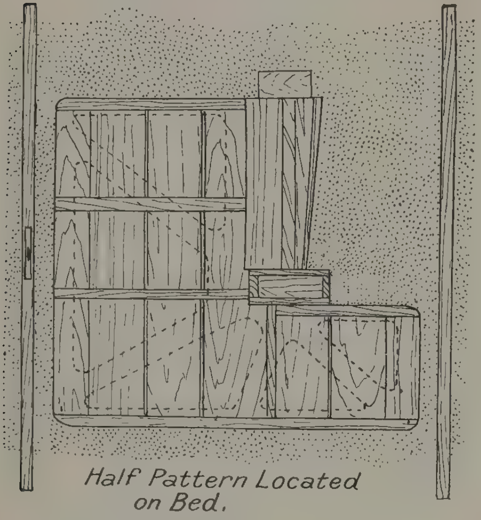
Sands suitable for cast iron are more plentiful, though few can be used without any admixture. When old loam is available from loam moulds, it is advantageously employed in conjunction with new sand in making suitable facing sand mixtures. Clay may be added to sands that are weak, while, on the other hand, a strong mixture may be weakened by the addition of blacking, sharp sand, or coke. No coal dust is added to dry sand facings for cast iron.

Somewhat similar mixtures are used for dry sand moulds to receive brasses or bronzes as for cast iron, although the texture is, as a rule, finer. With regard to dry sand moulds for aluminium, when these are employed, it is better to use a weaker mixture than would be used for other non-ferrous metals. Dry sand mixtures are preferably milled to toughen them.

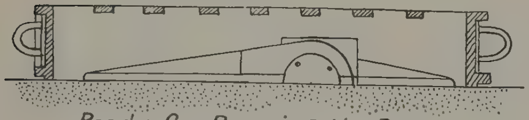
Surface Wash on Dry Sand Moulds. Castings made in dried moulds usually have a superior skin, and this is largely due to a refractory wash that is applied to the mould. The material used to form the wash varies with the metal; thus, for steel castings the moulds are painted with graphite and flour silica suitably bonded with china clay. For large work, "compo" is frequently thinned with water and used for the purpose. Mineral blacking is the main constituent of the wash for the moulds to receive cast iron. A carbonaceous material, like core gum, or clay, may be used as a bond. Graphite is generally included, as it improves the quality and facilitates sleeking. For bronzes or brasses, graphite is frequently used, especially for the former, but a combination of graphite and French chalk, or French chalk alone, gives a good surface. China clay, flour or peas meal may be used as a bond. French chalk is to be preferred for moulds to receive aluminium.



Preparing Sand Bed.



Half Pattern Located on Bed.



Ready for Ramming the Drag.

Fig. 20. Preparing Large Mould by Turnover Method

Turnover Method. Preparing moulds to be dried by the turnover process does not differ from that used in making green sand moulds, and, while preference is usually given to the bedding-in process, this does not mean that there is no place for the turnover method. Within limits it is profitable to make use of the turnover, providing suitable equipment is available. A level bed of sand is usually prepared on the foundry floor, and the pattern or pattern section laid upon it in an inverted position. The drag part is located about the pattern and on the prepared bed, additional bars being introduced in the box part to support the sand.

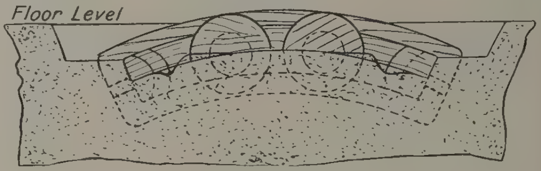


Fig. 21. Preparing a Drag in the Foundry Floor
Showing Mould Joint (Bedding-in)

Bedding-in Process. The expression, "bedding-in," is usually applied to that method of making a mould in which the bottom section, or, the pattern as a whole, is bedded into a previously prepared position in the sand floor. It may be applied more widely to the preparation of all moulds, for which a pattern is used, when the mould is formed in the position in which it is to be cast, and without the turning over of the drag. The process consists in making a hole, usually in the foundry floor, larger than the pattern to be bedded in, milled facing sand is then thrown over the bottom, and into this the pattern is forced until its surface coincides with the height at which it is desired to make a joint for the cope. With this method, some parts of the impression will be hard, while others will be soft; it is therefore necessary to remove the pattern in order to loosen and remove sand from hard places, and add sand to soft places, when the pattern is again bedded into its former position. The process is repeated

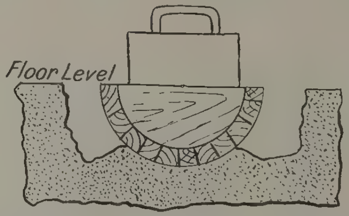


Fig. 22. Use of Foundry Floor for Bedding in Cylindrical Patterns

until the sand impression is of similar density, when the pattern is finally set and sand rammed about it to the joint surface. It is therefore necessary to bed in the pattern by easy stages so that regularity in the density of sand can be obtained.

This form of bedding-in is usually applied to the smaller class of work, whether in green or dry sand, for which boxes are not available for adopting the turnover method. There are two main alternatives to this form of bedding-in. Instead of the pattern section being hammered into the sand and removed, and the process continued until the sand is of regular density, a level surface or bed may be prepared on the foundry floor and a hole formed in it to accommodate the pattern. Facing sand is introduced to support the pattern so that its joint surface coincides with the prepared surface of the floor. Weights are used to hold the pattern in position, while sand is rammed about it. When the pattern or pattern section is strong, the prepared surface on the foundry floor is unnecessary, as the sand joint can be levelled off flush with the surface of the pattern. This method of bedding-in is only possible when there is sufficient room for ramming the sand after the pattern has been set, and the method is adopted for large cylindrical work when moulded horizontally. Preparing a level surface on the foundry floor is a common method when the pattern is large and comparatively frail, and when it is possible to secure battens to the surface of the pattern so that they rest on the sand surface. These reduce the possibility of twist, and thus maintain the shape of the pattern.

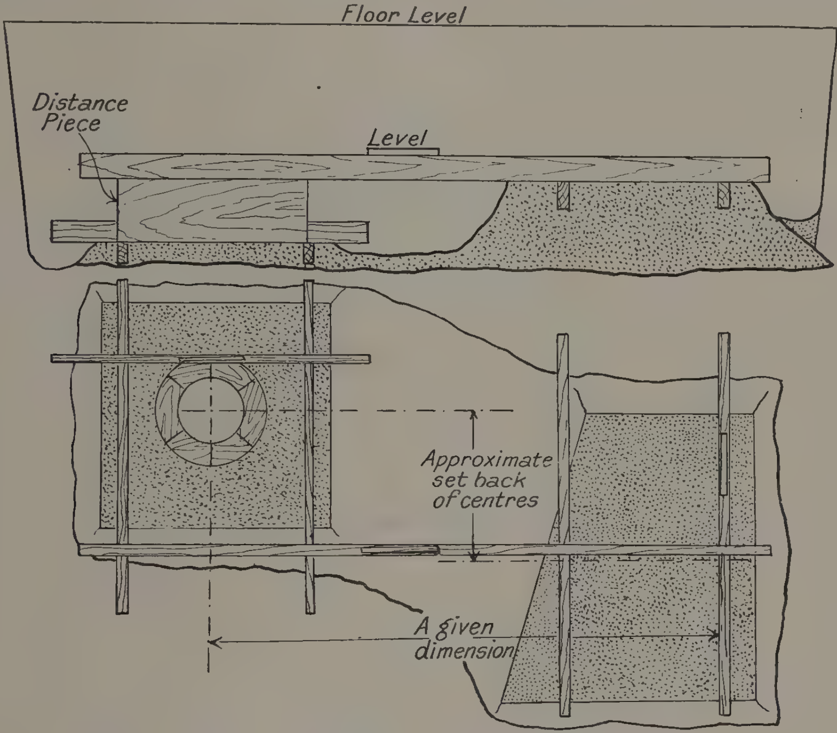


FIG. 23. PREPARATION OF BEARANCES TO RECEIVE PATTERN IN FOUNDRY FLOOR
Bedding-in

The other form of bedding-in is of special value when the pattern is large, and is comparatively flat on the surface which is to be cast down. There may be one or more flat surfaces, and the remainder accessible for ramming the sand against in the ordinary way. In such cases, a sand bed is prepared in a pit or case set in the foundry floor, and at a distance from the floor level that locates the pattern in the most suitable position for making the mould joint. In this way the sand forming the bed becomes part of the mould, and, apart from the greater ease in locating the pattern in the desired position, it possesses the advantage of being rammed readily to the required density before the pattern is introduced. When a flat surface on the pattern carries subsidiary projecting parts, the pattern is located on the prepared bed, so that an impression will be made by these projecting parts, and, on the pattern being removed, they are bedded in separately. When they have been set into the bed they should be withdrawn and fixed to the main pattern again, which is then located so that the impression will be filled. When the bottom of the pattern is not continuous and in the same horizontal plane, but consists of surfaces parallel to each other, a special bearance is required, upon which the different surfaces will rest.

Preparation of Bearances. Whatever form of bed is required, it is important that it should conform to the shape of the pattern, or, when the pattern is frail, or is badly constructed and has become twisted, to the shape the casting is required to have. An improperly or carelessly prepared bearance generally results in a distorted casting, because, owing to weights applied, patterns frequently follow the shape of the bed. Substantial straight-edged logs should be used and when bedded into the sand should be carefully tested with a spirit-level, both in the direction of their length and between each log. For the latter purpose, a parallel straight-edge is necessary, so that when it rests upon each log the spirit-level can be used on the top edge. When the work requires two or more bearances, in different planes, the bottom bearance should be prepared first, and a distance piece used for setting the next bearance. Bearances should be carefully checked before the logs are withdrawn, in case their positions have been altered with ramming. The impressions left by the logs are filled in with sand and levelled off flush with the bed.

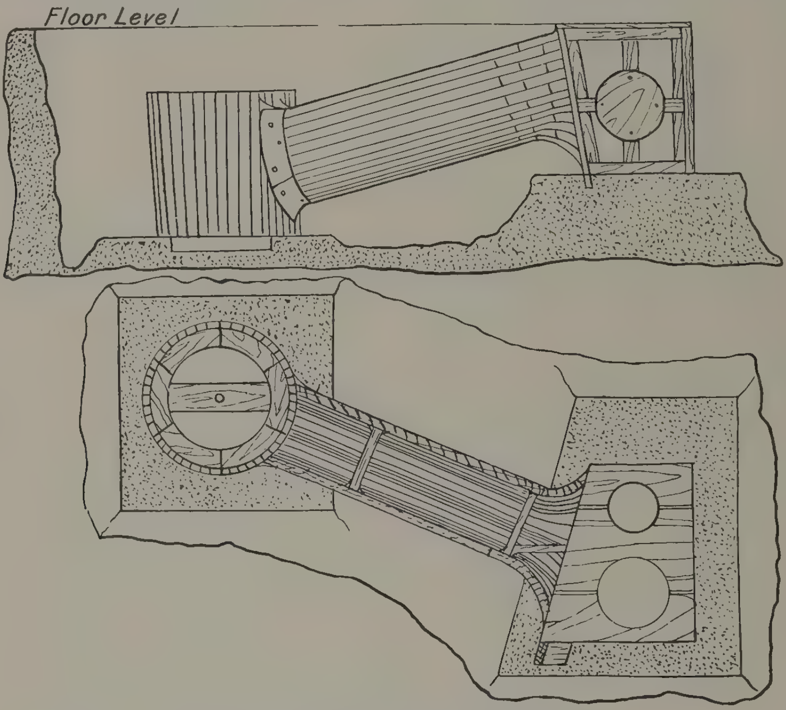


FIG. 24. PATTERN LOCATED ON BEARANCES
Ready for Preparation of Drag (Bedding-in)

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (Lond.)

LESSON VI

PLANT—(*contd.*)

Stove Cleaning. When using crude gas, the combustion chamber and checkers need frequent cleaning. This entails laying the stove off, opening all doors and allowing to cool, when stove cleaners enter the dome and drop chains or weights suspended on wires down each flue, thus removing the incrustations.

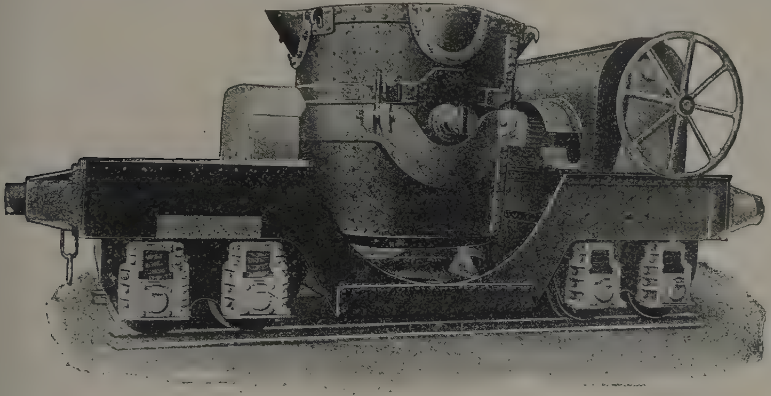


Fig. 21. Hot Metal Ladle

(Dewhurst)

It is generally found necessary occasionally to renew the brickwork in the combustion chamber opposite the burner, owing to the slagging effect of the hot dust. Dust can be shaken down the flues by discharging a stove gun under the checker arches.

The cleaning of gas has removed the necessity for cleaning and, by the use of smaller checkers, has enabled a much larger heating surface to be obtained in the same sized shell.

Heating Surface. A blast furnace stove usually has had an effective heating area of about 50,000 sq. ft., but modern practice is calling for greater areas. According to McLoughlin (Proc. Eng. Soc. W. Penna, 1926, 41, 420), stoves are now being built with areas of 90,000 to 100,000 sq. ft., equipped with pressure burners with automatic control of gas and air. These stoves have a radiation loss of about 8 per cent with an efficiency of about 80 per cent, and can deal with 60,000 cub. ft. of blast per minute to a straight line temperature of 1,400° F.

Cold Blast Mains from the engines to the stoves are designed to withstand the necessary blast pressure, and are constructed of steel plate. They are never lined, as they have no temperature to withstand. Blow-off valves are generally fitted so that the tubing is not strained beyond its safe limits.

HANDLING THE PRODUCTS

The Metal

Casting in Sand. In this method, an area in the vicinity of the front of the furnace is covered with sand, arranged with a slight fall away from the tapping hole. A trough or runner is made down the slope with branches

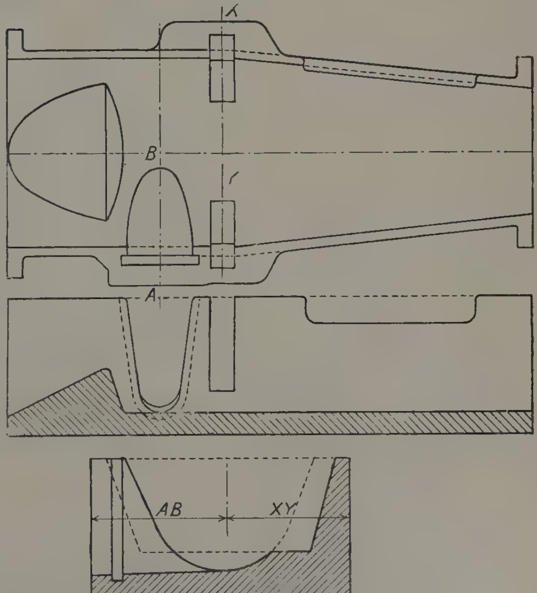


Fig. 22. Modified Killeen Skimmer and Dam

at right angles about 4 ft. apart. Along these runners are moulded, by means of wooden patterns, a series of parallel depressions.

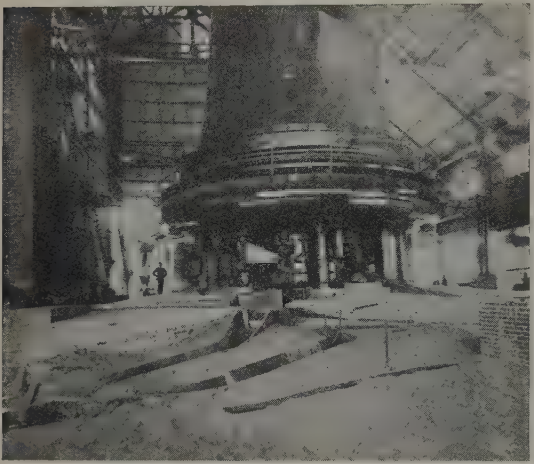


Fig. 23. View Showing General Arrangement of Cast House

The molten iron is run into these latter depressions first, generally, to the row most remote from the furnace. As the beds are filled, the neck of the runner is closed by means of a clay-grouted iron shutter. The combs of iron are allowed to cool, and are removed by means of an overhead crane to the pig breaker, or are broken up on the beds by manual labour. The latter operation is facilitated by throwing sand on the pigs, when they can be detached by means of iron bars and sledge hammers.

This process enables the iron to obtain an open-grained fracture, e.g. foundry irons. On the other hand, a certain amount of sand adheres to the pigs, which renders the iron rather unsuitable for steel making.

Casting in chills is a modification of the previous method, and consists in running iron into a permanent pig bed, made of cast iron. The latter is protected by washing with clay slurry or lime water. The iron is removed by spraying with water and lifting with bars. In this method open-grained iron is not obtained.

Casting Machines. In order to obviate the necessity for labour required to deal with large outputs, and to deal with week-end pig-iron in steel works, mechanical casting machines have been devised. The *Uehling* machine consists of two endless conveyors in tandem. Metal is poured into cast-iron troughs on the first and is solidified by being sprayed with water. The pigs are discharged on to the second belt constructed of steel plates, which carries them through a tank of water and dumps them at the end into trucks. The empty moulds on the first belt pass over sprays of lime wash on their return journey. The *Heyl* and *Patterson* machine performs both operations on the same continuous conveyor. The moulds pass slowly down an incline through a trough of water, emerge, and pass up a sharper incline to the dumping point. The empty moulds pass over sprays of lime water, or are coated with tar.

Casting in Ladles. As an economy in the manufacture of steel it is found advisable to send pig-iron in a molten condition to the steel furnace. The iron is tapped into metal ladles holding 15-60 tons of metal, and consisting essentially of heavy steel plated buckets lined with firebrick, and carried on a suitable carriage. The ladle is lifted and tilted by means of trunnions fixed to a belt, the metal being poured out by means of a spout. (See Fig. 21, showing a Dewhurst hot metal ladle.) Similar ladles are used in conjunction with casting machines, where the ladle is usually tilted by means of a worm gear or hydraulic ram

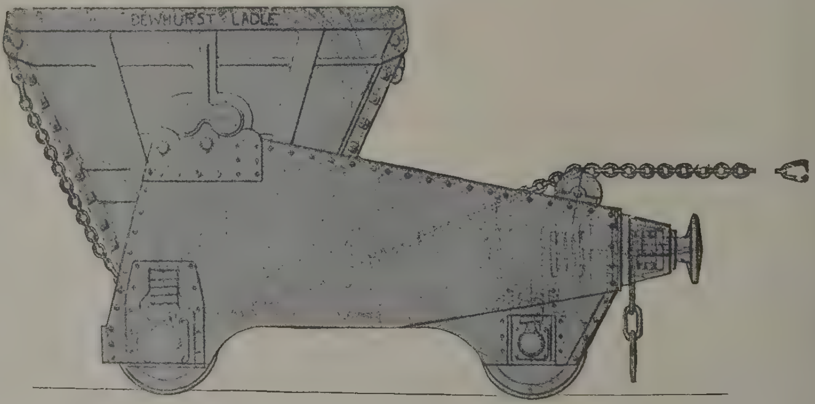


Fig. 24. Slag Ladle and Carriage
(Dewhurst

mechanism. Hot metal requires about 5 cub. ft. per ton, but an allowance in the depth should be made for skulling and to prevent overflowing on railway curves.

Iron Trough and Skimmer. Towards the end of the cast it is usual for a certain amount of slag to issue from the tapping hole on the top of the current of metal. This slag is removed by the skimmer and dam, the process consisting of fixing a skimmer across the metal trough in a vertical position until its lower edge nearly touches the surface of the molten iron. The latter is then caused to rise and flow over a dam in the bed of the runner, which, in conjunction with the skimmer forces the slag to overflow by a suitable sill at the side of the trough. (See Fig. 22 for a diagram of skimmer and dam.) The dam in old practice was made of sand, the skimmer being generally a steel plate, but with the advent of large outputs, the operation became rather hazardous, particularly if the sand was too damp and caused a "boil." In any case, with a sudden rush of metal the apparatus could not cope with the strain, the slag and iron passing freely down the beds together. To overcome this difficulty, *Killeen* devised a permanent apparatus, which consists of a cast-iron trough, shaped similarly to the old-fashioned trough and dam, and provided with suitable openings for slag removal and for draining the iron from the trough at the end of the cast. The whole is made on a generous scale in order to cope with sudden rushes of iron and slag. The skimmer is usually made of firebrick, set in an iron frame which can be raised at will. All metal work is protected with clay wash, the doors receiving a backing of sand. The general arrangement of a cast house is shown in Fig. 23, reproduced by permission of Messrs. H. A. Brassert & Co.

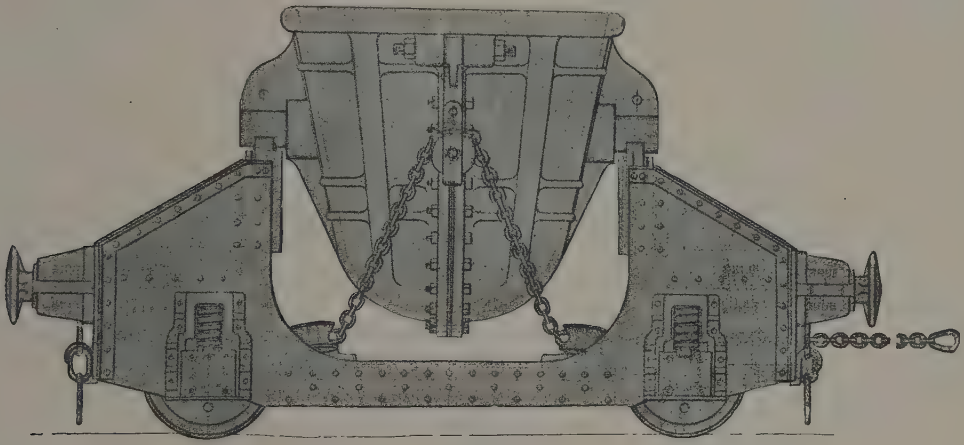


FIG. 25
Dewhurst)

SLAG

Tipping. The old method of tipping consisted in filling small pots with slag at the notch spout and allowing them to stand until the slag had solidified. The balls of slag were then dumped on a slag tip. It is now generally collected in larger ladles of the self-cleaning type holding 10-25 tons of molten slag. Carriages are designed so that ladles can be tipped sideways or endways, the latter type being useful in extending the tip and the former being used to fill in the space gained thereby. An extensively used type of slag ladle and carriage is that supplied by Messrs. Dewhurst, of Sheffield, and shown in Figs. 24 and 25, kindly supplied by that firm. The mechanism of the tilting operation is here clearly shown in both the side and end tipping ladles, which hold 15 tons of slag. The slag is generally tipped in a fluid condition, but is sometimes allowed to solidify for special purposes, e.g. road material. In other cases, the slag tip itself is cut away and broken down for the same purpose. Slag requires about 15 cub. ft. per ton.

Granulation. In this process the slag is allowed to emerge from a nozzle into a stream of water. The slag is solidified into small lumps or shot and, after draining in pits, is lifted out by grabs and used for ballast or concrete manufacture.

ENGINEERING SPECIFICATIONS

By Arthur P. Haslam, M.I.E.E.

LESSON V

TESTS FOR MATERIALS

It was pointed out in the first of these articles that the aim of the engineer should be to define such tests for the various materials he uses, that if samples on test pass these test conditions, it should be proof that the qualities of the material are uniform, and that when it is placed under similar conditions, the material will behave in a similar way.

The qualities of a material which are most important to the engineer are its tensile, shearing, torsional, and crushing strength, its elasticity, and its hardness. The materials in most common use are iron, steel, and cement, and it will be interesting to know how these materials are usually tested, and what are the requirements which should be met. Above and beyond the tests to which there is space here to refer, there are chemical analyses of composition, examinations of crystalline structure, and other most interesting ways of finding out the suitability of a substance for special uses.

Iron and Steel. Dealing very briefly with iron and steel, we find pure iron is difficult to obtain and has few commercial uses. It is fairly strong and can be hammered into shape. The nearest commercial material to pure iron is wrought iron, and the beautiful scroll ironwork produced by skilled craftsmen in many countries for several centuries shows what can be done with practically pure iron.

Molten iron has, however, the property of dissolving a small amount of carbon and of absorbing smaller amounts of sulphur, phosphorus, manganese, silicon, and tungsten, and the physical properties of the resultant material are considerably changed by small alterations in the amount of the varying substances present.

Iron is generally obtained from its ores by being heated to incandescence in a blast furnace, where coal or gas containing carbon is used in conjunction with a blast of hot air, both to heat the mass and reduce the ore to metallic iron. The product is termed pig-iron, which in addition to the iron, usually contains about 3 to 4·5 per cent of carbon, and a number of other metals and substances present as impurities.

Pig-iron remelted in a furnace in the presence of hot air loses part of its carbon, and we get cast iron containing about 2·5 or 3 per cent of carbon. Although the percentage of carbon may not be very different from that in pig-iron, the proportions actually combined with the iron and that present as minute particles or free carbon in the iron may vary.

Cast iron is comparatively hard, has, in good qualities, a tensile strength varying up to 10 or 12 tons per sq. in., and a compressive strength about four to four and a half times this amount. It becomes fluid at a relatively low temperature and can, with ease, be run into moulds and cast into any desired shape. These castings are hard and brittle, and a fracture shows a distinctly crystalline surface. It varies both in composition and physical qualities. The greater the carbon content, the harder, more brittle, and weaker the iron. The presence, in small quantities, of silicon tends to soften the iron, of manganese, to harden it, of phosphorus to render it more fluid when molten, and of sulphur to increase the proportions of combined carbon, and so to make the iron weaker and more brittle. It will be recognized that the manufacture of iron castings of a uniform quality demands the greatest care in using and mixing the correct grades of pig-iron.

If iron castings are packed in such substances as black oxide of manganese, red hematite ore or black hammer scale, and are then heated at red cherry heat for several days and very gradually cooled, the qualities of the castings are in some respects changed. They are much softer, less brittle, and can be more easily magnetized than before. In this form the iron is termed malleable iron, and such castings have an increasing commercial value.

Intermediate between wrought and cast iron, in carbon content, come the great family of steels, the study of which has been the life work of many eminent men.

Speaking generally, steels are compounds of iron in which carbon is present in quantities varying from 0·1 to 1·0 per cent, and silicon, manganese, sulphur, and phosphorus in varying small proportions. The additions of small quantities of nickel, chronium, and tungsten to steel, previously carefully prepared and purified of certain other elements, greatly varies, and in important directions improves the qualities of the steel. The general aim of the steelmaker is to reduce the percentage of sulphur and phosphorus to small definite limits, and by controlling the proportion of carbon content to obtain a steel of definite and uniform physical qualities.

In general, increasing the amount of carbon in steel increases its hardness and tensile strength, while it reduces the percentage elongation obtainable when stretched in a testing machine.

The qualities of finished steels depend to a large extent on the method of manufacture. Pig-iron, the intermediate product between the ore and the finished steel, contains a much higher percentage of carbon, and pig-iron can be converted into steel, either by taking from it in some way the extra carbon, or by first of all taking out all the carbon and then adding to the iron sufficient carbon to make the required steel. The Bessemer process of steel making is based upon the first principle, the open hearth process on the second. The best steels are obtained by the second process, as it is not easy in the Bessemer process to eliminate other elements. For certain uses of steel it is usual to specify, in addition to the limits of tensile strength and elongation, that it shall be made by the open hearth process.

Cement. Portland cement is another article to which great attention has been paid by engineers, as upon its uniform quality depends the stability of much of our building work and the safety of our foundations. Like steel, it has a varying composition, and its essential qualities may differ considerably if the greatest care is not taken in its manufacture. The B.E.S.A. early gave close attention to its standardization, and several successive revisions of the original specification have resulted in very nearly all the Portland cement manufactured in this country complying with its requirements.

It is made from a mixture of chalk and clay ground together in a wet state, which mixture is heated to a clinkering temperature and the resultant material is ground to a very fine powder.

The finished product should consist of approximately 22 per cent of silica, 62 per cent of lime, 5 per cent of alumina, 2·75 per cent of sulphur content, 2·5 per cent of oxide of iron, and 3 per cent of magnesia.

Cement Testing. The tests usually applied concern the chemical composition, the principal requirements being that the lime content shall not exceed 62 per cent; the specific gravity which should not be less than 3·10; the fineness; the time taken for setting and the tensile strength. The latter is taken both for neat cement, and for a mixture, in specified proportions, with sand of a certain quality.

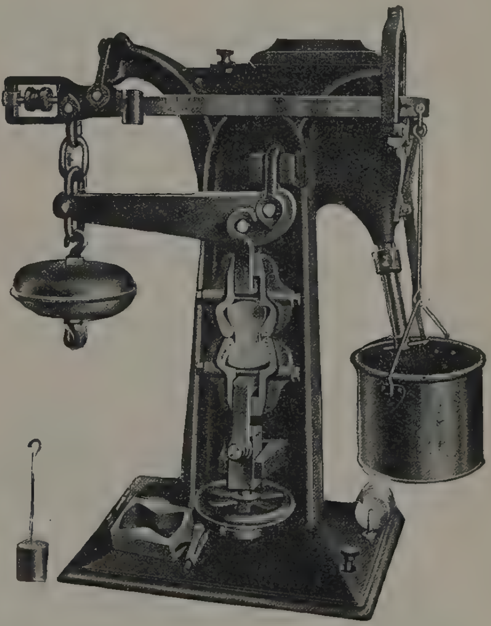


FIG. 1. DENISON CEMENT-TESTING MACHINE

These tensile tests consist in making small briquettes of a given size which have at their testing point an area of 1 sq. in. The briquettes are first kept in the moulds in which they are made in a damp atmosphere for 24 hours. They are then taken from the moulds and placed in fresh water. They are tested at the end of 7 and of 28 days for breaking strength. At the end of 7 days this should not be less than 450 lb. for neat cement, and 200 lb. for the mixture of cement and sand, and at the end of 28 days this should have gradually increased in accordance with a definite formula.

There are several machines made for carrying out this test. One of the most popular is illustrated in Fig. 1, and is made by Messrs. Samuel Denison & Son, Ltd. It acts on the double-lever principle. The sample briquette is placed in position, while wet, between the holding jaws of the machine, and pressure is gradually applied by means of small lead shot flowing into a bucket hanging at the end of the steelyard. As soon as the pressure is sufficient to fracture the briquette, the supply of lead shot is automatically cut off. The breaking strain is ascertained by unhooking the bucket and hanging it at the other end of the steelyard, when it is weighed in the ordinary way, the weight of the shot indicating the stress in lb. per sq. in.

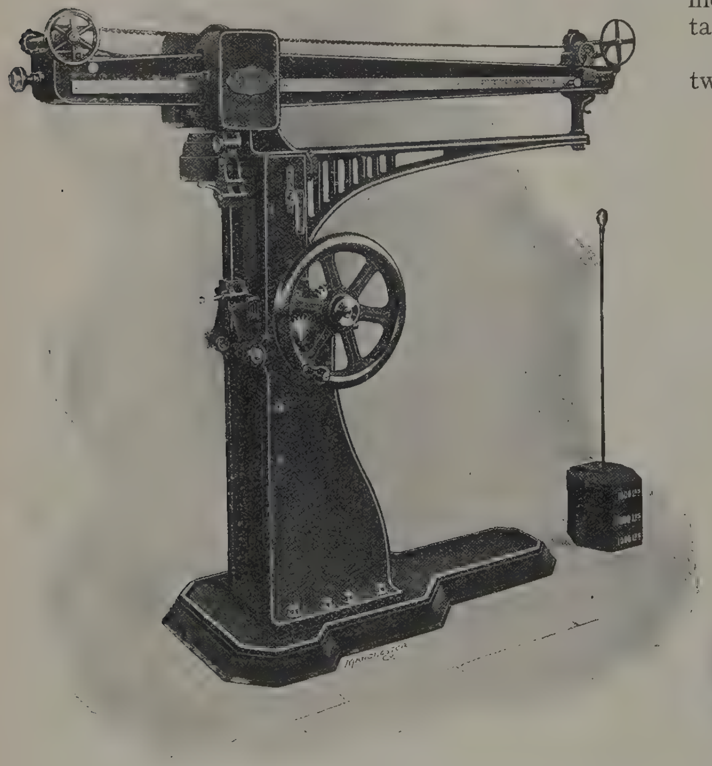


Fig. 2. Denison Wire-testing Machine

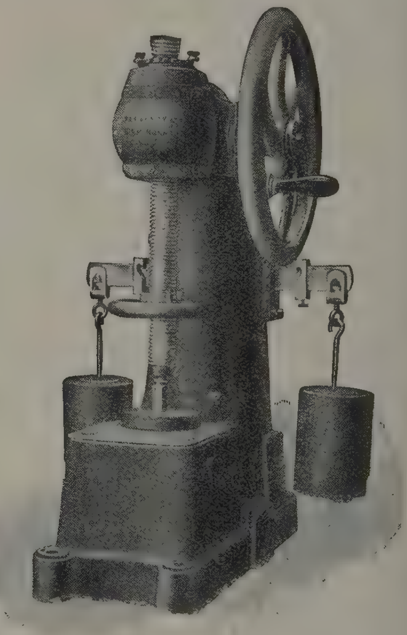


Fig. 3. "Brinell" Testing Machine

Testing Machines for Iron and Steel. The machines which are used for testing iron and steel samples are in most cases very accurate, powerful, and expensive. There are a number of types in successful use, and for descriptions of these, the reader is referred to the Lessons on "Engineering Materials, Testing and Properties," and also on "Applied Mechanics."

For wire testing, a machine made in various sizes by Messrs. Samuel Denison & Son, Ltd., is often used. It is illustrated in Fig. 2, and will test up to a load of 8,000 lb. It weighs and indicates the exact amount of stress applied, and automatically indicates the extension which has taken place in the test pieces.

The test length of wire is held between special grips, and the strain is applied by means of a hand-wheel working through worm gear, which gives a steady gradual increase of the stress. As will be seen, the machine acts on the single-lever principle, the movement of the poise weight along the beam applying the strain and registering the amount.

It is often desirable to take a test for the hardness of a substance and what is called the "Brinell" test is generally made. This consists of the application of a standard load of 3,000 kg. to a steel ball 10 mm. in diameter, in order to press it into the surface of the material to be tested.

The load is applied for a period of a quarter of a minute, after which the resulting impression is measured by a microscope, and the diameter in millimetres is referred to a table of "Brinell" hardness numerals. In this way the hardness of the material under test is determined. Various machines have been designed for making this test. Fig. 3 shows one made by Messrs. W. & T. Avery, Ltd., of Birmingham, which is widely used. The test specimen is placed on the table, and the steel ball is adjusted into position by turning the setting wheel on the screw. The load is then applied by turning the large hand-wheel, the moment of application of the full-load being indicated by the rise of the steelyard weights. The application of the test is simple, and with ordinary care it is not easy to exceed the prescribed load. Many other machines for special tests have been made and are in constant use.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XIII

THE RISE OF TEXTILE ENGINEERING

SIR RICHARD ARKWRIGHT (1732–1792)

Since the dawn of history, the textile industry has ever proved to be one of the most practically important occupations of mankind. But, in spite of this fact, the great spinning and weaving industries of the world have hardly been occupying their present gigantic status for more than a hundred and fifty years.

During the early portion of the eighteenth century, cotton spinning and weaving were carried on in very much the same manner as they were in biblical times. The primitive spinning wheel and the hand loom were then the sole means of producing textile fabrics, and the industry still continued to be a typically rural one, the necessary operations being, for the most part, carried out in country cottages during the long evenings and winter days when purely agricultural pursuits were not possible.

The present cotton industry of this country, and, indeed, of the world, had its rise in the invention by Kay, of Bury, of the "Flying Shuttle," a device by means of which the shuttle in the loom was carried mechanically from side to side, instead of being laboriously pushed by hand. This invention, in itself, increased the output of woven cloth tremendously.

The subject of our present notice, Sir Richard Arkwright, was probably the greatest pioneer of the textile industry. His invention of the "Spinning Jenny" so facilitated the production of yarn that the industry was immediately placed on a large scale manufacturing basis, a

Sir Richard Arkwright

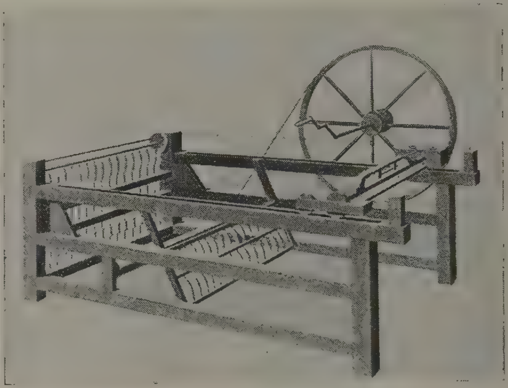
position from which it has progressively advanced ever since.

Born in Preston, Lancashire, in 1732, Richard Arkwright was the thirteenth child of very poor parents. He received little or no education, and, during his early manhood, he carried on the trade of a barber surgeon. It is on record that Arkwright during these early days invented a process for dyeing human hair, but whether the invention was successful in practice it is impossible to say.

The Spinning Wheel

Arkwright married at an early age, and immediately after his marriage, his thoughts turned to the construction of some device whereby the operations necessary to the process of spinning could be greatly facilitated and increased in speed. The inventor spent many patient years over this project. Patient they were because his wife, with doubtful zeal, took care to destroy as many of her husband's models as she could lay hands on, in the hope of bringing his activities back to the more profitable pursuits of barber and blood-letter.

Eventually, however, Arkwright perfected his mechanical spinning machine, which he called a "spinning jenny." The invention was a practical success. In its essentials, it consisted of a number of spindles mechanically linked together in such a manner that fifteen to twenty times the amount of yarn could be spun in the same amount of time, and with a less expenditure of energy as was the case with the time honoured spinning wheel.



Arkwright's "Spinning Jenny"

At first, the spinning jenny was hand operated. Afterwards, Arkwright used the power of horses for working his machine, and finally he employed water-power. So successful was the invention of the spinning jenny, that Arkwright rose to considerable fame, and he amassed a very respectable fortune. He received the honour of knighthood at the hands of George III, and, when death finally brought a close to his career in 1792, he had the no uncertain pleasure of knowing that his invention had revolutionized the whole of the textile industry, and had conferred a benefit on humanity, the limits of which were at that time impossible to conceive.

SIR ROBERT A. HADFIELD, Bt.
D.SC., D.MET., F.R.S., F.I.C., M.INST.C.E.

Vice-President Inst.Mech.E.; President of the Iron and Steel Institute, 1905–1907; President of the Faraday Society, 1914–1920; Master Cutler of Sheffield, 1899–1900
Bessemer Gold Medallist, 1904; John Fritz Gold Medallist, 1921; Howard Quinquennial Prizeman, 1902. Telford and Stephenson Medals; Elliott-Cresson Gold Medal; Three Gold Medals of the Société d'Encouragement pour l'Industrie Nationale

SPECIAL STEELS

By Sir Robert Hadfield, Bt., D.Sc., D.Met., F.R.S., F.I.C., M.Inst.C.E.
Membre Correspondant de l'Academie des Sciences
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PART I

INTRODUCTION

It is well known that ordinary steel is an alloy of iron and carbon containing less than 1·5 per cent of carbon, and more or less manganese up to about 1 per cent. The term "Special Steels" is therefore used here to designate those steels which owe their characteristic properties to one or more alloying elements other than carbon. Such steels are also known as "Alloy Steels." Ordinary steels in contra-distinction are often referred to as "Carbon Steels."

When one considers the variety of elements now used in the composition of alloy steels, and the wide range afforded by the addition of one or more of them, each in varying proportions, it is seen how great is the variety of properties which is possible in this field.

The elements principally employed for alloying, apart from carbon, are manganese, chromium, nickel, silicon, tungsten, and vanadium, and to a lesser extent cobalt, molybdenum, and copper.

Aluminium and titanium also have their uses in steel manufacture, not so much as truly alloying elements as in facilitating metallurgical processes. For this purpose they are not added in such quantity as to leave any appreciable amount in the finished steel.

The element carbon profoundly affects the properties of iron even within the limit just mentioned, namely, about 1·5 per cent, modifying its melting point, strength, hardness, magnetic, electrical, and other qualities, according to its amount, conferring upon it too the valuable property of becoming intensely hard by sudden cooling from high temperatures.

In a general way it may be said that when carbon steel containing about ·40 per cent of carbon and upwards is heated to a red heat, that is 750° C. or above, and quenched in water, the material is found to be much harder than before. On the other hand, by reheating and cooling slowly, a hardened carbon steel can be softened again. On these well-known facts is based the science of "heat treatment." The development of means of accurate measurement of high temperatures has enabled the metallurgist to control with certainty the production of desired qualities within the capacity of the material, by suitable rates and degrees of heating and cooling. It has been found that these effects are largely due to modifications, physical and chemical, in the relation of carbon to iron.

THE BENEFICIAL EFFECTS OF ALLOYING ELEMENTS UPON IRON

With these facts in mind, a useful conception of alloy steels is obtained by regarding the function of the alloying elements as affecting the relations between the carbon and the iron. In practice, this operates in many cases by providing the desired qualities by means of a simpler heat treatment, with consequently increased certainty of achieving the result desired, owing to the greater ease of control.

In the case of carbon steels, it is found that even with an elaborate heat treatment, a given desirable mechanical property may in general be obtained only by a sacrifice of some other feature. Thus, if a high tensile strength be obtained, the ductility and resistance to shocks will be greatly impaired, and vice versa. Suitable alloy steels provide these properties in combination with each other, and herein mainly lies their advantage for stressed parts.

The alloying element, in addition to affecting the relations between the carbon and the iron, often, however, has what may be regarded as a direct effect of its own, and thus alloy steels yield a much greater range of mechanical properties than is afforded by simple carbon steels; in many cases, too, they possess exceptional physical properties of value. Whereas commercially pure iron (99·9 per cent iron) has a tensile strength of 18 to 20 tons per sq. in., and a ductility represented by 40 per cent extension, high tenacity alloy steels are available which have tensile strengths exceeding 100 tons with a ductility of as much as 15 per cent, or, when in the form of wire, a tenacity of even 200 tons. One of the nickel-manganese alloy steels prepared by the author showed an extension of 76 per cent on an 8 in. test piece, with a tenacity of nearly 60 tons per sq. in. Another steel discovered by the author, and known as "manganese steel," has a tenacity of 73 tons per sq. in., accompanied by as much as 73 per cent of elongation. There is no other material extant which possesses such a combination of qualities to the same extraordinary degree.

Alloy steels, in addition to placing at the disposal of the engineer materials having qualities quite unattainable with carbon steel, can often be substituted with advantage even in those cases where carbon steel has the necessary qualities. This is specially so where weight is of importance, as, for example, the use of alloy steels resulting in a material reduction of weight with an equal margin of safety.

Such advantage may, in fact, operate in a double way. The stresses which moving parts have to sustain are of two kinds. There are those which it is their function to transmit or support, and which are more or less under control. On the other hand, there are the dynamic stresses, due solely to the weight and inertia of such moving parts. These dynamic stresses serve no useful purpose, but on the contrary, in high speed machinery, may easily become excessive, due to lack of proper balancing or to the inability of heavy moving parts to accelerate or retard their speed, in response to sudden fluctuations in load, whether inseparable from the working conditions, or due to inexpert driving. Many actual breakdowns occur from this cause.

Whatever other considerations there may be, therefore, for reducing weight, unnecessary weight in moving parts may be positively harmful. By permitting a lightening of the sections, high tenacity alloy steel acts to its own advantage in reducing the stresses which it has to support. Thus, the reduction in weight which can be effected is greater than would appear at first sight. At the same time design achieves a greater degree of exactitude, and performance greater reliability, by the risks from the effects of inertia becoming minimized. Clumsiness, too, gives way to neatness.

In some cases where ordinary steel may be otherwise satisfactory, the wastage, whether due to wear, corrosion or scaling under the effects of heat, is excessive. This involves such frequent replacements, with consequent heavy expenditure both in material and labour, not to mention inconvenient delays and sometimes loss of revenue, that the use of alloy steels at a higher initial cost is often a true economy, due to their much greater durability. These and other considerations help to explain how it is that the use of alloy steels has made so much headway in modern engineering.

TYPES OF ALLOY STEELS AND THEIR PRACTICAL APPLICATIONS

The manner in which the requirements of the engineer have been met by the discovery and development of suitable alloy steels is, perhaps, best demonstrated by the following description, necessarily brief, and with the help of illustrations, of some of the more important alloy steels and their applications.

(*a*) **Manganese Steel.** It was in 1882 that the author commenced his experiments on the effect of manganese on the properties of iron, leading to the discovery of the material known as manganese steel. The results were published in his first papers, read before the Institution of Civil Engineers in 1888, "Manganese in its Application to Metallurgy," and "Some Newly-discovered Properties of Iron and Manganese." In the first paper an account is given of the mechanical properties of a series of steels containing from 0·83 to 21·69 per cent of manganese, a range much more extensive than any which had been the subject of previously published researches. The second paper contained particulars of the extraordinary physical properties of these materials, and particularly of manganese steel.

The name "manganese steel," refers to an alloy containing about 1¼ per cent of carbon and about 13 per cent of manganese. It was the first alloy steel, that is containing an alloying element in considerable proportions, to be applied to practical purposes. The most striking features of manganese steel may be summarized as follows—

(*a*) Manganese steel is practically non-magnetic, notwithstanding the fact that it contains about 86 per cent of the magnetic metal iron.

(*b*) The alloy, instead of being hardened and made comparatively brittle, as is the case with carbon steel, is rendered softer and tougher by quenching.

(*c*) It has high tensile strength, 60 to 73 tons per sq. in. when suitably heat treated, combined with an extraordinary elongation, viz., 50 to 73 per cent, exceeding that obtainable with the purest iron, having a tenacity of only 19 tons.

(*d*) A remarkable propensity for hardening up to as high as 600 Brinell hardness under the influence of deformation. As the result, manganese steel is extraordinarily resistant to abrasion, and the more so the more severe the service to which it is applied. Its practical non-machinability, that is, the resistance it offers to the action of cutting tools, is due to the same cause.

Manganese steel when molten is very fluid and hence may readily be cast into moulds of intricate shape, making greatly for the convenience of its application to many purposes. The castings are remarkably free from blow-holes. As regards rolling, experience in the control of the material and of its heat treatment, has made it possible to produce successfully not only sheets and plates, but also rails from which complicated assemblies, such as layouts, including points and crossings for railways and tramways may be built up. These are made also in the form of castings.

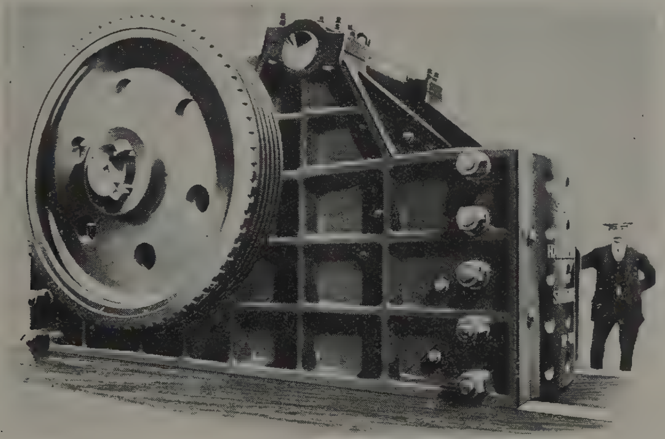

FIG. 1. JAW CRUSHER

Size 54 in. by 36 in. Weight 90 tons, for crushing hard rock to 6 in. or 8 in. cubes, with an output of 150 to 200 tons per hour. All the wearing parts are of manganese steel

The remarkable characteristic of resistance to wear by abrasion makes manganese steel pre-eminently suitable for the jaws of stone and ore crushing machines, dredger pins, bushes, lips, and other parts, and, in fact, for numerous applications where duty of this nature has to be performed. It is a remarkable fact that the highest wear resisting qualities are developed only under the hardest working conditions. In the various operations of machining, the action of the tool hardens manganese steel so much as to make it unworkable. Thus, manganese steel is practically unmachinable, and in order to give it a smooth surface resort is made to the operation of grinding.

These various mechanical operations have been worked out in a satisfactory manner at the works of the author's firm, Messrs. Hadfields, Ltd., Sheffield, where the original discovery of manganese steel, with its many applications for industrial requirements, occurred.

(*b*) **Silicon Steel.** While at work upon the study of manganese alloys the author was also investigating the effect of additions of silicon to steel, and the results of this research were embodied in his paper read in 1889 before the Iron and Steel Institute, "On Alloys of Iron and Silicon." The range of silicon in the series was from 0·24 to 8·83 per cent. It was there shown that the opinion, commonly held at the time, as to the harmful effects of silicon in steel was ill-founded. The author was able to invent and produce in considerable quantities tool steel containing high percentages of silicon possessing very superior cutting qualities. Later, high tenacity steel containing silicon to the amount of about 1 per cent was successfully used for ship plates. Another application of the alloys of iron and silicon in rather higher percentages, namely, 15 to 20 per cent, is referred to in connection with non-corrodible steels. A considerable reference will also be found to these various applications of silicon steel, in the author's recent book, *Metallurgy and Its Influence on Modern Progress*, to which those interested in this important branch of science are referred.

One of the important applications of silicon steel is for electrical engineering. The successful use of silicon steel for the purpose beforementioned, that is tool steel, had been achieved with the presence at the same time of an appreciable percentage of carbon, and the author of this paper decided that it was desirable to ascertain what influence silicon had upon iron in the absence of other elements. For this purpose he carefully prepared an extensive series of alloys of iron and silicon kept as free as possible from carbon, which was present only in very small amounts. In a joint research on their electrical and magnetic qualities by the late Sir William Barrett, F.R.S., Mr. W. Brown, and the author, and fully described in their joint papers published in the proceedings of the Royal Dublin Society, 1899–1904, and the Institution of Electrical Engineers, 1902, it was found that a steel containing about 3 to 4 per cent of silicon when suitably heat-treated by the methods indicated in the various patents of the author, has higher magnetic permeability, lower hysteresis loss, and lower eddy current losses than pure iron.

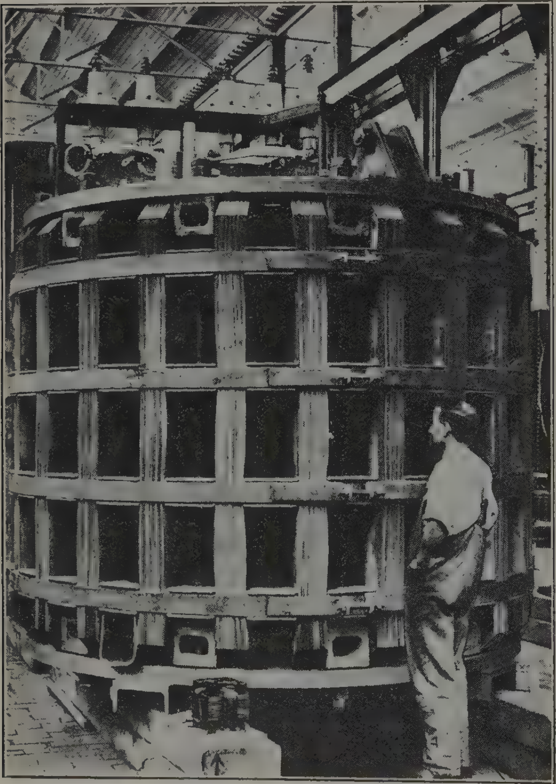

FIG. 2. SILICON STEEL CORE OF A MODERN TRANSFORMER OF 6,000 k.V.A. CAPACITY

The first transformer, of 0·5 k.V.A. capacity, constructed from silicon steel in 1903, is also shown by way of contrast

In the case of the materials, mostly iron or very mild steel, used for transformers and dynamos at that time, the magnetizing losses in addition to being very high originally, often, through what is termed "ageing," doubled or even quadrupled after a few months' service. This necessitated the expensive dismantling and re-annealing of the transformer plates to avoid excessive wastage of electrical energy. With the first transformers constructed from silicon steel it was found that the losses, initially about two-thirds that of charcoal iron, actually slightly *decrease* during years of service. Its non-ageing character has since been amply confirmed by experience, which has also enabled the material to be still further improved in its low hysteresis qualities.

First produced in practical quantities in 1906, and since achieving world wide use, it is estimated that silicon steel is now effecting a saving to the world of 18 million pounds annually, representing the fuel equivalent of the energy losses averted. The total savings effected to date since its introduction probably amount to not far short of 160 million pounds sterling. When, further, the great assistance which it must have given to the development of the electrical industry is considered, that is in enabling electrical energy to be supplied efficiently and economically, the description of silicon steel, by the eminent authority, Professor J. A. Fleming, F.R.S., as "an immensely valuable material" will be understood.

Other types of special steel investigated by the author have been described in his various papers including, "Alloys of Iron and Aluminium," 1890; "Alloys of Iron and Chromium," 1892; "Alloys of Iron and Nickel," 1899; "Alloys of Iron and Tungsten," 1903. Some of these represent the first systematic researches carried out on alloys of their types. Further papers by the author contain accounts of his researches on alloy steels containing more than one special element.

(*To be continued*)

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
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LESSON XIV

GEOMETRY—(*contd.*)

The Circle. We need not define or describe a circle, but we may emphasize that all its properties are due to the fact that every point on the circumference is at the same distance from the centre. Consequently, for instance, every triangle formed by two radii and a chord is isosceles (see Fig. 41, triangle OAB). Hence a perpendicular from the centre on a chord bisects the chord, and, conversely, a line joining the centre to the mid-point of the chord is perpendicular to the chord, i.e. OD in the Figure is perpendicular to AB and also bisects AB.

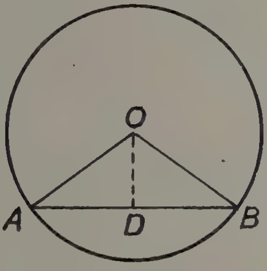


FIG. 41

We are thus able to calculate lengths of chords, etc., by use of the right-angled triangle rule. If the chord is 2 in. distant from the centre in a circle of radius 5 in. (in Fig. 41, $OA = 5$ in. $= OB$, $OD = 2$ in.), we can find the length of the chord. For

$$AD^2 = OA^2 - OD^2 = 25 - 4 = 21$$
$$AD = \sqrt{21} = 4{\cdot}583 \text{ in.}$$
so that $$AB = 2\,AD = \underline{9{\cdot}166 \text{ in.}}$$

Another instance of the use of the above fact is seen if we know the span and rise of a circular arch, and require the radius of the circle. If ACB (Fig. 42) is the arch, the span AB is 20 ft., and the rise CD 6 ft., what is the radius of the circle? Completing the circle, and calling the centre O, and the radius r, we have $OA = r = OC$, so that $OD = r - 6$; also $AD = \frac{1}{2} AB = 10$,

Now $$OA^2 = OD^2 + AD^2 \qquad \text{(1)}$$
i.e. $$r^2 = (r-6)^2 + 10^2 = r^2 - 12r + 36 + 100$$
whence $12r = 136$,
so $$r = 11\tfrac{1}{3} \text{ ft.} = \underline{11 \text{ ft. } 4 \text{ in.}}$$

This case is sufficiently important for us to obtain a formula for it. Calling the height of the segment, CD, h, then $OD = r - h$, so, if $AD = c\ (= DB)$, we have, by (1) above,

$$r^2 = (r-h)^2 + c^2 = r^2 - 2rh + h^2 + c^2$$
Transposing, $2rh - h^2 = c^2$,
or, factorizing, $\underline{h(2r - h) = c^2}$.

This is the best form in which to remember it, for the second factor on the left is the length of a line in the figure, being that of ED, where CDE is the diameter through C. It is possible, then, to write the formula in terms of lines in the figure, thus

$$CD.DE = AD^2 = AD.DB$$

In the latter form we have a particular case of an important property of the circle, that if two chords intersect, the product of the segments of one is equal to the product of the segments of the other. For the present, we state the fact without proof. In Fig. 43, (*a*) and (*b*), we have

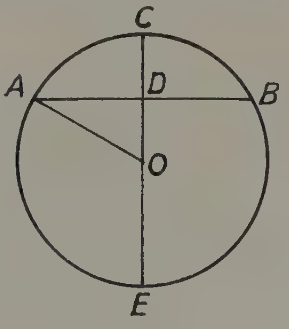


FIG. 42

$$AX.XB = CX.XD.$$

In (*b*), imagining CD to rotate about X until it becomes a tangent, XT, each of these products must also be equal to the square on XT.

Angle Properties. Important and useful as are the above chord properties of a circle, there are properties of angles connected with a circle that are more striking, and equally important. Thus, in Fig. 44, if A and B are two points on the circumference, and O is the centre, the angle ACB, C being another point on the circumference, is always *half* the angle AOB. To see that this must be true, join CO and produce it to D. Then triangle AOC is isosceles, so that $\angle OAC = \angle OCA$. Also $\angle DOA$, being the exterior angle of the triangle AOC, is equal to $\angle OAC. + \angle OCA$ Therefore

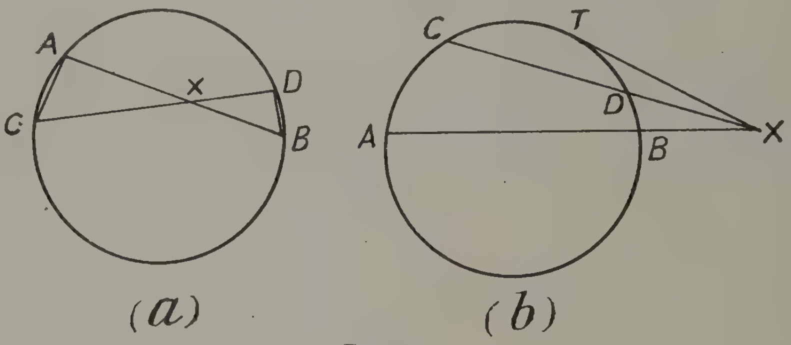


FIG. 43

$$\angle DOA = 2\angle OCA$$

Similarly, $\angle DOB = 2\angle OCB$

Adding, $\angle DOA + \angle DOB = 2\angle OCA + 2\angle OCB$

or $\angle AOB = 2\angle ACB.$

This is evidently true for all positions of C, a proof on similar lines holding for the cases in Fig. 45, where, in the second case, it is the *reflex* angle AOB which is twice the angle ACB. The result of this is that in Fig. 46, all the marked angles are equal. That is: angles at the circumference of a circle, standing on the same arc, and on the same side of it, are equal, each being half the angle at the centre standing on the same arc.

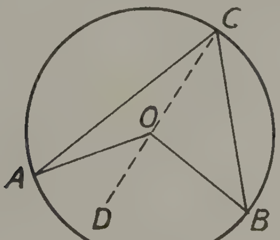


FIG. 44

A very important particular instance is that of Fig. 45(c), where A and B are the extremities of a diameter. Then, $\angle AOB = 180°$, so that $\angle ACB = 90°$. Consequently, **the angle in a semicircle is a right angle.** This fact is the basis upon which several geometrical constructions requiring perpendiculars rest.

Another consequence is found in considering the angles of a cyclic (i.e. inscribed in a circle) quadrilateral, such as $ABCD$ in Fig. 47. There, by the above,

$$\angle ADC = \tfrac{1}{2}\angle AOC$$

$$\angle ABC = \tfrac{1}{2}\text{reflex }\angle AOC,$$

so that $\angle ABC + \angle ADC = \frac{1}{2}$ (sum of angles at O) $= \frac{1}{2}$(4 right angles) $=$ 2 right angles.

Thus, the sum of the opposite angles of a cyclic quadrilateral is two right angles (or 180°).

Tangent. A line which *touches* a circle is called a *tangent*. It is usually better to consider tangent from the point of view of a chord that has moved gradually away from the centre, and is just on the point of leaving the circle altogether, as indicated in Fig. 48. The points of intersection P, Q, come gradually closer,

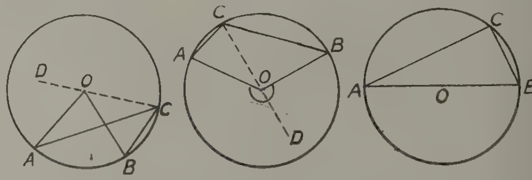


FIG. 45

until they ultimately coincide at T. During this process, the line joining the mid-point R to the centre is always perpendicular to the chord, so that, in the limiting position, when P, Q, and R all coincide at T, we must still have OT perpendicular to the tangent. Thus: any tangent to a circle, and the radius through the point of contact, are at right angles.

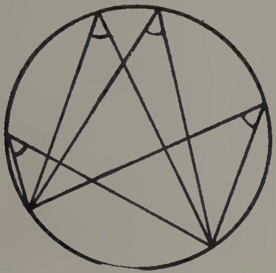


FIG. 46

Mensuration of Circle. The circumference of a circle is not *exactly* expressible in terms of the radius or diameter, though the ratio $\frac{\text{circumference}}{\text{diameter}}$ is the same for all circles. The value of this important ratio, always denoted by the Greek letter π, and its value can be calculated to as many places as we desire. $3\frac{1}{7}$, $3{\cdot}142$, and $3{\cdot}1416$, are the values, progressively more accurate, used in engineering calculations. (It is preferable, in numerical work, to keep the symbol π until the actual calculation is to be done, and then choose the value which gives results as accurately as are needed.)

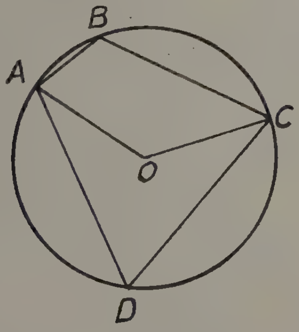


FIG. 47

Thus, if C is the circumference, d the diameter, and r the radius of a circle, we have

$$C = \pi d = 2\pi r \text{ (since } d = 2r\text{)}.$$

We can find the length of an *arc* of a circle by simple proportion if we know the angle subtended at the centre, for the whole circumference subtends 360° there. For instance, if the radius of the circle is 6 in., and the angle subtended is 55°, the length of the arc is

$$\frac{55}{360} \times 2\pi \times 6 = \underline{5{\cdot}76 \text{ in. approximate.}}$$

Radian Measure of Angles. If we denote the angle by A, measured in degrees, then the length of the arc is

$$\frac{A}{360} \times 2\pi r = \frac{\pi A}{180}\, r$$

If we have many dealings with arcs, it will be convenient to have the factor $\frac{\pi A}{180}$ tabulated for values of A. This is done, and the values are termed the radian, or circular measure of the angles. Thus, if an angle is $A°$, it is also θ radians, where

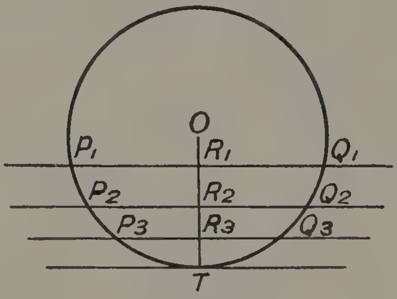


FIG. 48

$$\theta = \frac{\pi A}{180}$$

and then we have the very simple formula for the arc,

$$\text{arc} = r\theta.$$

Another way of looking at this is to put $\theta = 1$, that is, find the unit in this new way of measuring angles. If we do so, we get

$$\text{arc} = \text{radius (when angle} = 1 \text{ radian)}.$$

Thus the unit angle is one in which the arc is equal to the radius—and this is evidently the same angle in all circles. It is, moreover, a *natural* way of selecting a unit angle, as distinct from dividing a right angle into a quite arbitrary number, such as 90, equal parts. In advanced work we shall see that radian measure is, on account of its naturalness, very much used. For conversion remember that

$$\pi \text{ radians} = 180°.$$

Area. If we divide a circle into a very large number of sectors by radii, as in Fig. 49, we can treat each tiny sector as a triangle, with base the small arc, and height the radius, so that its area is $\frac{1}{2}$ arc × radius. For a number of these, the total area will be $\frac{1}{2}$ (sum of arcs) × radius, that is

$$\begin{aligned}\text{area of sector} &= \tfrac{1}{2}\text{ arc} \times \text{radius},\\ &= \tfrac{1}{2}r\,\theta \times r = \tfrac{1}{2}r^2\theta,\end{aligned}$$

where θ is the radian measure of the angle at the centre. For the complete circle, the area is

$$\begin{aligned}\tfrac{1}{2}\text{ circumference} \times \text{radius} &= \tfrac{1}{2} \times 2\pi r \times r\\ &= \pi r^2 = \tfrac{1}{4}\pi d^2.\end{aligned}$$

The area of a segment of a circle can be obtained by subtracting the area of the triangle from that of the sector, but we shall refer to this later, when studying trigonometry.

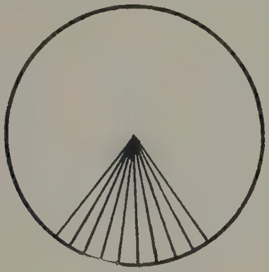


FIG. 49

EXERCISE No. 24

1. A circular shaft 2 in. in diameter is flattened to take a key, the greatest thickness of the cut being $\frac{1}{3}$ in. Find the width of the flat.
2. The span of an arch is 15 ft., and the rise 6 ft. Find the radius of the circle. By drawing to scale find the angle at the centre, and calculate the length of the curve.
3. Three circles, each 1 in. in diameter, are such that each touches the other two. Find the area of the space in the middle.
4. Find the radian measure of 45°, 60°, 81°, and 204°. Also the number of degrees in 1, 0·3, and 1·195 radians.
5. The wheel of a bicycle has a diameter of 28 in. How many times will it turn in going a mile ?
6. Convert into radians per second, 130 r.p.m., and 3,000 r.p.m.

Similar Figures. In drawing to scale, in maps and plans, in photography, and in projection by a magic lantern or kinematograph, we deal with figures of the same shape, but of different size. From our experience we know that in such cases, we copy the angles, but we reduce or magnify the lengths in a definite ratio. Pairs of figures of the same shape, but of different size, are said to be *similar*, the word "similar" used in geometry being a technical term meaning a much more definite "likeness" than when used in ordinary conversation. In fact, to say that two geometrical figures are similar, means two very definite things.

1. All angles of one figure are equal to the corresponding angles of the other.
2. All lengths of one figure bear the same ratio to the corresponding lengths in the other.

More briefly,

1. **Corresponding angles equal ;**
2. **Corresponding lengths proportional.**

In the case of triangles, however, these things can be shown to depend upon one another ; two equiangular triangles *must* have their sides proportional ; and if two triangles have their sides proportional, they *must* be equiangular. Space will not permit going into this very fully, but we will deduce a few consequences of these facts.

Earlier in this lesson, we stated that when two chords of a circle intersect, the product of the segments of one is equal to the product of the segments of the other. In Fig. 43(a) above, join AC and BD. Then $\angle A = \angle D$, both standing on the arc CB ; and for a like reason $\angle C = \angle B$. Thus, the triangles AXC and DXB must be equiangular, and consequently have corresponding sides proportional. Hence,

$$\frac{AX}{XD} = \frac{CX}{XB} \text{ whence, cross-multiplying,}$$

$$\underline{AX.XB = CX.XD.}$$

ANSWERS TO EXERCISES NOS. 22 AND 23

(22)

2. $+5°$; 106°, 95°, 143°, 72°, 124°.
3. 140°.
4. 20.
5. $\angle A = \angle B$, $\therefore$ $AE = BE$; since also $AD = BC$, $ED = EC$, $\therefore \angle EDC = \angle ECD$, $\therefore \angle ADC = \angle BCD$.
7. Join AD. Then triangle ABD = triangle DCA (two sides and included $\angle$), etc.

(23)

1. and 4. 1·984 sq. in., 10·39 sq. in., 21 sq. cm.
2. 18·0 ft. (approx.).
3. 13·0 cm., 5·686 cm.,

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XIV

BEAM DEFLECTION

MATHEMATICALLY AND GRAPHICALLY

In the previous lesson we have derived the expressions for the deflections of beams supported and loaded in different ways. A worked example will now be given to illustrate the method of solving, both mathematically and graphically, the case of a beam simply supported at its ends and carrying a number of loads. It is assumed that I and E are constant.

Example. A beam spans 24 ft. and carries 3 loads of 2, 4, and 3 tons at distances of 4 ft., 10 ft., and 18 ft. respectively from the left-hand support. Determine the deflection under each of the loads, the maximum deflection, and where it occurs.

$E = 13{,}500$ tons per sq. in., and $I = 10{,}000$ (in.)^4 units.

We will take the left-hand support A as the origin, and consider each length AB, BC, CD, and DE of the beam separately.

Length AB

On any section aa, whose distance from the origin is x, the B.M.

$$= M = EI\,\frac{dy^2}{dx^2} = 4\cdot75x_1$$

$$\therefore \quad EI\,\frac{dy}{dx} = \frac{4\cdot75x_1^2}{2} + F \quad . \quad . \quad (1)$$

$$\text{and} \quad EI\,y = \frac{4\cdot75x_1^3}{6} + Fx_1 + G \quad . \quad (2)$$

Length BC

On any section bb, the B.M.

$$= M = EI\,\frac{d^2y}{dx^2} = 4\cdot75x_2 - 2\,(x_2 - 4)$$

$$\therefore \quad EI\,\frac{dy}{dx} = \frac{4\cdot75x_2^2}{2} - (x_2 - 4)^2 + H \quad . \quad (3)$$

$$\text{and} \quad EIy = \frac{4\cdot75x_2^3}{6} - \frac{(x_2-4)^3}{3} + Hx_2 + J. \quad . \quad (4)$$

Length CD

On any section cc, the B.M.

$$= M = EI\,\frac{d^2y}{dx^2} = 4\cdot75x_3 - 2\,(x_3 - 4) - 4\,(x_3 - 10)$$

$$\therefore \quad EI\,\frac{dy}{dx} = \frac{4\cdot75x_3^2}{2} - \frac{2(x_3 - 4)^2}{2} - \frac{4(x_3 - 10)^2}{2} + K \quad . \quad . \quad (5)$$

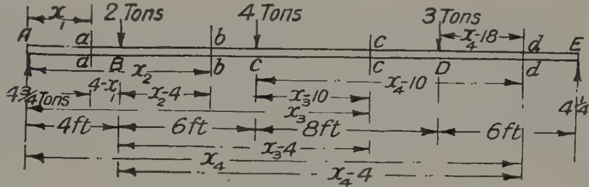


Fig. 73

$$\text{and} \quad EIy = \frac{4\cdot75x_3^3}{6} - \frac{(x_3 - 4)^3}{3} - \frac{4(x_3 - 10)^3}{6} + Kx_3 + L \quad . \quad (6)$$

Length DE

On any section dd, the B.M.

$$= M = EI\,\frac{d^2y}{dx^2} = 4\cdot75x_4 - 2(x_4 - 4) - 4(x_4 - 10) - 3(x_1 - 18)$$

$$\therefore \quad EI\,\frac{dy}{dx} = \frac{4\cdot75x_4^2}{2} - \frac{2(x_4 - 4)^2}{2} - \frac{4(x_4 - 10)^2}{2} - \frac{3(x_4 - 18)^2}{2} + N. \quad . \quad (7)$$

$$\text{and} \quad EIy = \frac{4\cdot75x_4^3}{6} - \frac{(x_4 - 4)^3}{3} - \frac{4(x_4 - 10)^3}{6} - \frac{3(x_4 - 18)^3}{6} + Nx_4 + P. \quad . \quad (8)$$

Our next step is to evaluate the constants and this is possible since we know certain end conditions. The point B may be considered as being either the end of the length AB *or* the beginning of the length BC, and hence, the deflection of B, as given by (2) = the deflection of B as given by (4); also, the slope of the beam at B as given by (1) = the slope of the beam at B as given by (3).

Similar considerations apply to the points C and D.

Hence, when $x = 4$; (1) = (3), that is

$$\frac{4{\cdot}75x_1^2}{2} + F = \frac{4{\cdot}75x_2^2}{2} - (x_2 - 4)^2 + H.$$

Substituting $x = 4$ in the above equation, we find that $F = H$.

Also, when $x = 4$; (2) = (4), that is,

$$\frac{4{\cdot}75x_1^3}{6} + Fx_1 + G = \frac{4{\cdot}75x_2^3}{6} - \frac{(x_2 - 4)^3}{3} + Hx + J.$$

Substituting $x = 4$ in the above equation, and remembering that $F = H$, we find that $G = J$.

Similarly, we may deal with the deflection and slope at the point C, when $x = 10$, (3) = (5) and (4) = (6), and we find that $H = K$ and $J = L$.

Also, with reference to point D, when $x = 18$, (5) = (7) and (6) = (8), and we find that $K = N$ and $L = P$.

From the above we have, $F = H = K = N$
and $G = J = L = P$.

A further condition of which we make use is that, when $x = 0$, $y = 0$; and substituting these values in (2), we find that $G = 0$; ∴ also J, L, and $P = 0$, and we have disposed of four of our constants. Again, when $x = 24$ in (8), $y = 0$.

Substituting these values in (8) we obtain

$$0 = \left[\frac{4{\cdot}75 \times (24)^3}{6} - \frac{(20)^3}{3} - \frac{4 \times (14)^3}{6} - \frac{3(6)^3}{6} + N24\right]$$

$$\therefore\ N = -\frac{1585}{6} = K = H = F.$$

Having now determined the numerical value of the constants, we can proceed to calculate the deflections under the several loads, and the position and magnitude of the maximum deflection, by the aid of the appropriate equations.

Deflection at B

Substituting $x = 4$ in either (2) or (4), we have

$$y = \frac{1}{EI}\left[\frac{4{\cdot}75 \times 64}{6} - \frac{4 \times 1585}{6}\right] = -\frac{1006}{EI}\text{ ft.}$$

Deflection at C

Substituting $x = 10$ in either (4) or (6), we have

$$y = \frac{1}{EI}\left[\frac{4{\cdot}75 \times 1000}{6} - \frac{216}{3} - \frac{15{,}850}{6}\right]$$

$$= -\frac{1922}{EI}\text{ ft.}$$

Deflection at D

Substituting $x = 18$ in either (6) or (8), we have

$$y = \frac{1}{EI}\left[\frac{4{\cdot}75 \times (18)^3}{6} - \frac{(14)^3}{3} - \frac{4(8)^3}{6} - \frac{1585 \times 18}{6}\right]$$

$$= -\frac{1394}{EI}\text{ ft.}$$

The maximum deflection will occur at some point on the beam between C and D, and at this point the beam will be horizontal, that is, $\frac{dy}{dx} = 0$; hence we can use (5) and write

$$\frac{dy}{dx} = 0 = \frac{4{\cdot}75x_3^2}{2} - (x_3 - 4)^2 - 2(x_3 - 10)^2 - \frac{1585}{6}$$

the solution of which gives $x = 11{\cdot}83$ ft.

The magnitude of the maximum deflection is obtained by substituting $x = 11{\cdot}83$ in (6), and we may write

$$y_{max} = \frac{1}{EI}\left[\frac{4{\cdot}75 \times (11{\cdot}83)^3}{6} - \frac{(11{\cdot}83 - 4)^3}{3} - \frac{4(1{\cdot}83)^3}{6} - \frac{11{\cdot}83 \times 1585}{6}\right]$$

$$y_{max} = -\frac{1978}{EI}\text{ ft.}$$

It should be noted that the deflections are obtained in feet, it being more convenient to work in feet units, since the numbers involved are smaller. The values of E and of I, however, are given in inch units, so that a change of units is required here. This is effected as follows—

$$E = \frac{13{,}500\text{ tons}}{\text{in.} \times \text{in.}} = \frac{13{,}500\text{ tons} \times 12\text{ in.} \times 12\text{ in.}}{\text{in.} \times \text{in.} \times \text{ft.} \times \text{ft.}}$$

$$E = \frac{13{,}500\text{ tons} \times 144}{\text{ft.} \times \text{ft.}}$$

$$E = 1{,}944{,}000\text{ tons per sq. ft.}$$

$$I = 10{,}000\ (\text{in.})^4 = 10{,}000\ (\text{in.})^4 \times \left(\frac{(\text{ft.})}{12\text{ in.}}\right)^4$$

$$I = \frac{10{,}000}{(12)^4} \times (\text{ft.})^4 = 0{\cdot}4822\ (\text{ft.})^4.$$

Thus, the deflection at B

$$= -\frac{1006}{EI}\ \text{ft.}$$

$$= \frac{-1006}{1{,}944{,}000 \times 0{\cdot}4822}$$

$$= 0{\cdot}00107\ \text{ft.}$$

$$= -0{\cdot}0129\ \text{in.}$$

The other deflections may be obtained in like manner.

GRAPHICAL METHOD

The deflection diagram for a loaded beam of constant E and I can easily be drawn by the following method. We will take the same beam and loading as in the previous example, and draw the deflection diagram. Set down the beam and load positions to any linear scale, say, 1 in. to 4 ft. Using a convenient first load scale of, say, 1 in. to 2 tons, set down the line of loads, *abcd*. Fix any pole P at a convenient polar distance, $x =$ say, 3 in. Draw the first funicular polygon or B.M. diagram in the usual way (Fig. 74).

The B.M. scale = linear scale × load scale × first polar distance.

$$\text{,,} \quad \text{,,} \quad = \frac{4\ \text{ft.}}{\text{in.}} \times \frac{2\ \text{tons}}{\text{in.}} \times 3\ \text{in.}$$

$$\text{,,} \quad \text{,,} \quad = \frac{24\ \text{tons-ft.}}{\text{in.}}$$

This B.M. diagram is now to be used as a load diagram, the loads being proportional to the several areas into which the whole B.M. diagram is divided, and, further, the loads are taken as acting through the CG of each of the separate areas. Dividing the B.M. diagram into triangles, we find by measurement that $a_1 = \frac{19}{48}$ sq. in.,

$$a_2 = \frac{19}{32}\ \text{sq. in.},\ a_3 = \frac{35{\cdot}5}{32}\ \text{sq. in.},\ a_4 = \frac{35{\cdot}5}{24}\ \text{sq. in.}$$

$$a_5 = \frac{25{\cdot}5}{24}\ \text{sq. in., and}\ a_6 = \frac{25{\cdot}5}{32}\ \text{sq. in.}$$

A second load scale must now be selected and used in setting down these areas, a_1, a_2, a_3, a_4, a_5, and a_6 as loads on a new load line. A convenient scale will be 1 in. to 1 sq. in. of area of B.M. diagram. These areas, a_1, a_2, etc., are shown set down on the second line of loads *e*, *f*, *g*, *h*, *i*, *j*, and *k*. Choose a second polar distance, y, say, 4 in., and complete the second funicular polygon in the usual method. The deflection curve is that which is tangential to the sides of the polygon.

Deflection Scale

The B.M. represented by each ordinate of the B.M. diagram is proportional to (1) the first linear scale, (2) the first load scale, and (3) the first polar distance x; hence, the B.M. scale which is used to measure ordinates on this diagram, is the product of the above three factors, or

B.M. scale = first linear scale × first load scale × first polar distance.

The areas, a_1, a_2, etc., depend upon the B.M. scale, and the first linear scale; these areas, considered as loads are set down on the second line of loads to the second load scale. Further, the width FG of the deflection diagram depends upon the second linear scale employed, and, for convenience, this is the same as the first linear scale. The ordinate of the deflection diagram also depends upon the second polar distance y. From the above it follows that the deflection scale ∝ (first linear scale × first load scale × first polar distance × first linear scale × second load scale × second linear scale × second polar distance).

and since deflection $\propto \frac{1}{EI}$ we have

Deflection scale $= \frac{1}{EI}$ (first linear scale × first load scale × first polar distance × first linear scale × second load scale × second linear scale × second polar distance), and, in our particular example,

$$\text{Deflection scale} = \frac{1}{EI}\left[\frac{48\ \text{in.}}{\text{in.}} \times \frac{2\ \text{tons}}{\text{in.}} \times 3\ \text{in.} \times \frac{48\ \text{in.}}{\text{in.}} \times \frac{1\ \text{sq. in.}}{\text{in.}} \times \frac{48\ \text{in.}}{\text{in.}} \times 4\ \text{in.}\right]$$

and inserting the given values for E and I we have

$$\text{Deflection scale} = \frac{24 \times (48)^3\ \text{tons (in.)}}{13{,}500 \times 10{,}000\ \text{tons (in.)}}$$

$$= 0{\cdot}01966\,;$$

that is to say, each inch of ordinate of the deflection diagram indicates an actual beam deflection of 0·01966 in.

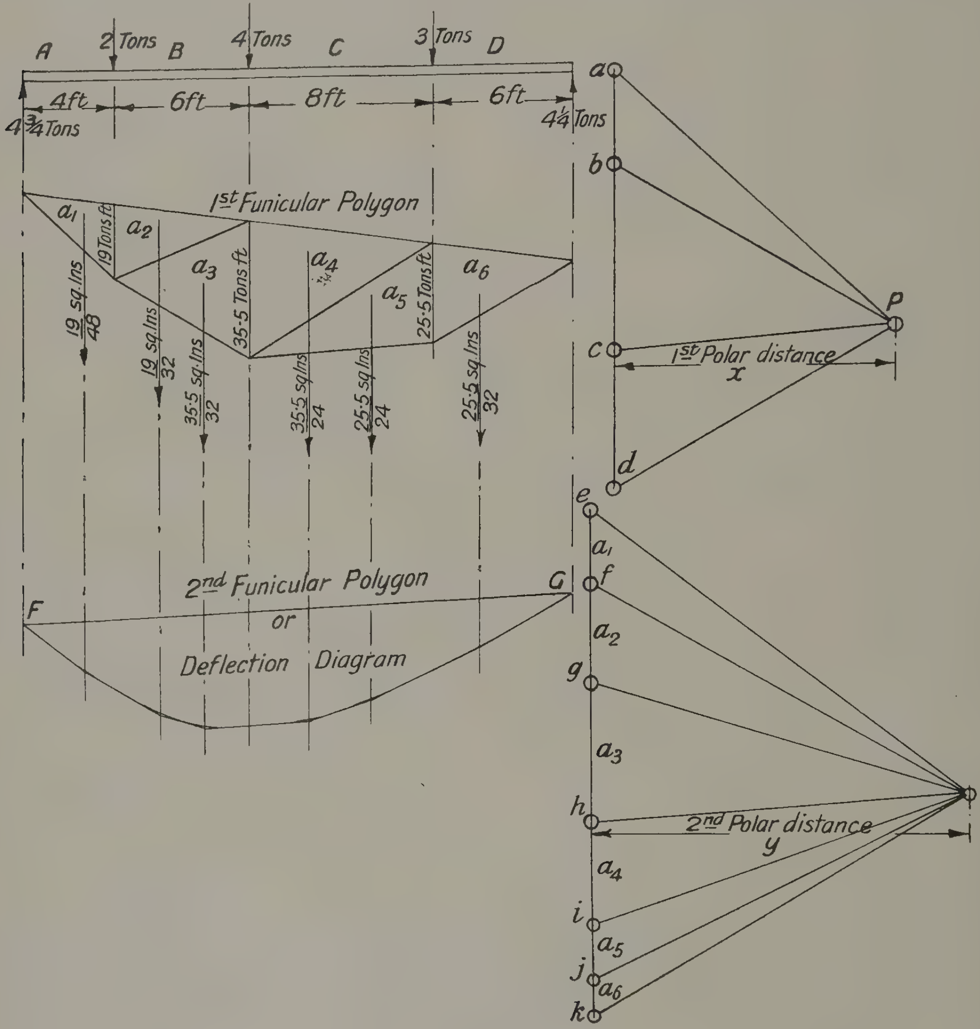


FIG. 74

In confirmation of the graphical method we have used above, we note that, if we integrate the *load* on one side of a section of a beam, we obtain the *shearing force*, F, and again, if we integrate the shearing force we obtain the bending moment, M; or

$$\int w \, . \, dx = F \; ; \text{ and } \int dF . \, dx = M.$$

Now, drawing the B.M. diagram is the graphical equivalent of the above double integration; or $\iint w \, . \, dx \, . \, dx = M$.

Similarly, $\int \frac{M}{EI} \, . \, dx = \text{slope}$, and $\int \text{slope} = \text{deflection}$

$$\therefore \iint \frac{M}{EI} \, dx \, . \, dx = \text{deflection.}$$

Drawing the deflection diagram by derivation from the B.M. diagram is the graphical equivalent of the above double integration.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XIII

Choice of Type. As we proceed to review the art of machine construction and drawing we are led to observe that it consists, in a large measure, of a process of building with old bricks. This does not imply that initiative and ingenuity find no place in the art, but rather that the good designer is one who can correctly ascertain the value of the component machine part and, at the same time, arrange the components in the most economical and efficient manner.

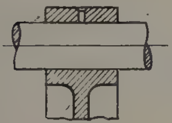


Fig. 1

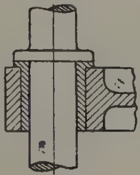


Fig. 2A

Analysing most machines, we find that each part may be found to exist in other machines. Moreover, some machine parts are very common and, as a consequence, have the greater number of variations in existing design. We find most machines possessing one or more journal bearings, and we may take the choice of a journal bearing as a good example in the method of choosing a type.

TABLE I

Type	Load	Service	Remarks
1	Light	Slow and irregular	Wear negligible. Suitable for brass bearing brackets
2	Light	Slow but regular	Wear moderate. Suitable for cast-iron bearing brackets
3	Light	Regular moderate speed of rotation	Wear small due to rings maintaining an oil film on journal
4	Heavy	Continuous, but moderate peripheral speed	Antifriction metal, which is comparatively soft, prevents local overloading in case of malalignment
5	Heavy	Continuous and high peripheral speed	Forced lubrication necessary to provide an oil flow sufficient to take the heat generated in friction
6	Moderate	Continuous or irregular high or low speeds	Efficient and easily lubricated

Figs. 1–6 show various types of journal bearings, and Table I gives a brief summary of their respective merits and utility.

Application. We may now proceed to use the above references in finding a suitable type of journal bearing to meet the following specification—

Diameter of journal = 6 in.
Speed ,, ,, = 100 r.p.m.
Bearing load = 3000 lb.
continuously vertical.

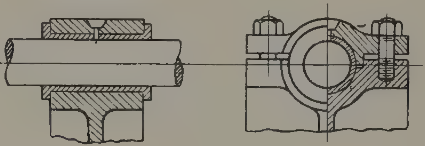


Fig. 2B

We observe the load to be heavy and pass on to the next point in the classification *service*.

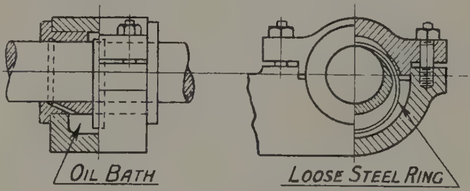


Fig. 3

First, finding the peripheral speed of the journal,

let v = speed in ft. per sec.
D = diameter of journal in in.
N = speed of journal in r.p.m.

$$\text{then } v = \frac{\pi DN}{12 \times 60} = \frac{DN}{230} = \frac{6 \times 100}{230} = 2{\cdot}6.$$

This speed is moderate and, the nature of the loading being continuous, we may choose Fig. 4 as a suitable type of bearing, providing the bearing surface (when fixed) is large enough to radiate the heat generated in friction.

The load being applied always in the one direction, the bearing pressure ought not to exceed 50 lb. per sq. in. of projected surface. Allowing this pressure we find the length of bearing

$$l = \frac{3000}{50 \times 6} = 10 \text{ in.}$$

Assuming a coefficient of friction $\mu = \cdot 02$. Then the work done in overcoming frictional resistance

$$= \text{bearing load in lb.} \times \mu \times v \text{ ft.-lb. per sec.}$$

$$= 3000 \times \cdot 02 \times 2\cdot 6 = 156 \text{ ft.-lb. per sec.}$$

and the heat generated

$$= \frac{\text{work done in ft.-lb./sec.} \times 3600}{778} \text{ B.Th.U.'s hour.}$$

$$= \frac{156 \times 3600}{778} = 722 \text{ B.Th.U.'s per hour.}$$

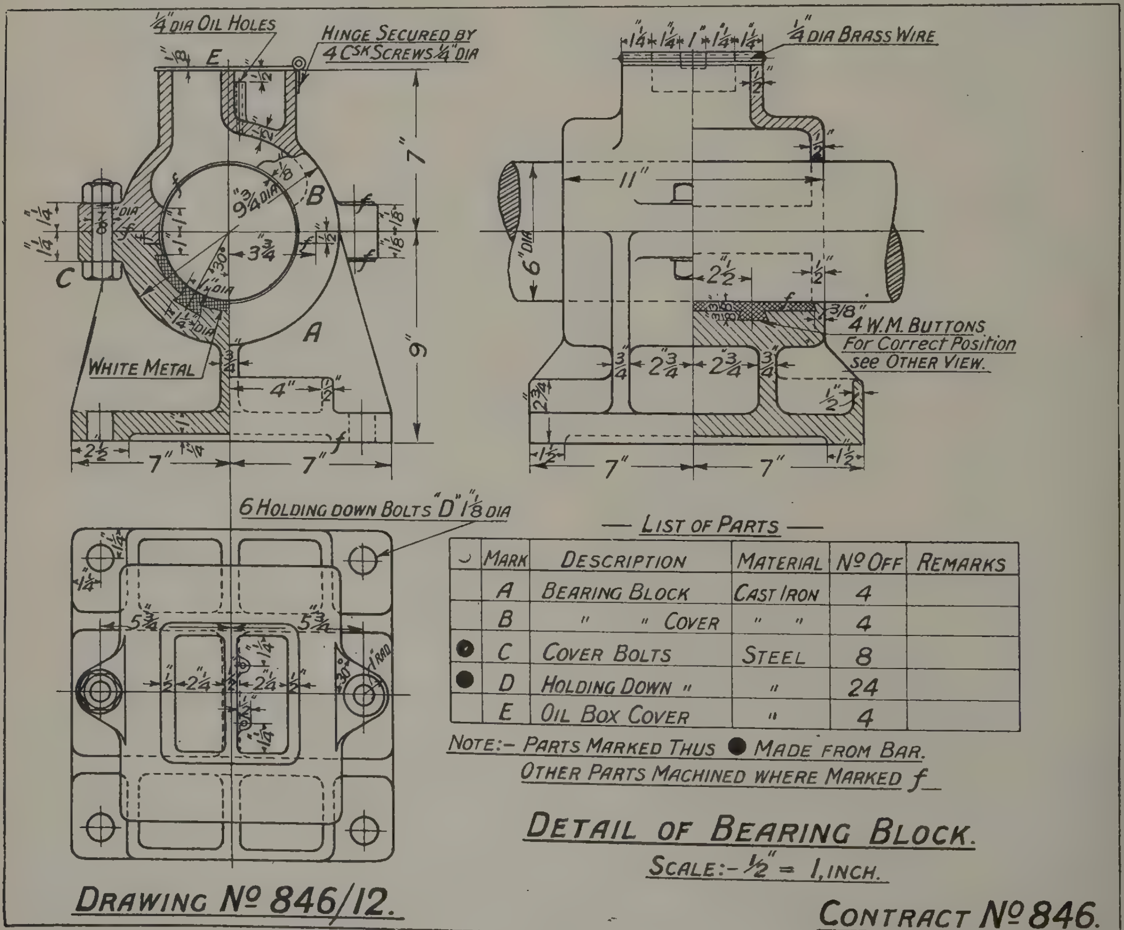
We may allow a value of 60 B.Th.U.'s per hour for each sq. in. of projected bearing surface, as an average value of heat radiation for bearings in naturally ventilated spaces.

In the above bearing the heat generated per sq. in. of surface

$$= \frac{722}{6 \times 10} = 12\cdot 03$$

which value is well below the maximum quoted above.

High Speed Journal. As an alternative example, take the case of a steam turbine journal running at 3,000 r.p.m., the bearing load being 3 tons and the journal diameter 6 in.



¼" DIA OIL HOLES
HINGE SECURED BY 4 CSK SCREWS ¼" DIA
E
B
A
C
9¾" DIA
3¾
WHITE METAL
7"
9"
4"
2½
7"
7"
¼" DIA BRASS WIRE
11"
6" DIA
2½
2¾
2¾
3/8"
4 W.M. BUTTONS For Correct Position see OTHER VIEW.
6 HOLDING DOWN BOLTS "D" 1⅛" DIA
5¾
5¾
2¼
2¼
1" RAD

— LIST OF PARTS —

	MARK	DESCRIPTION	MATERIAL	Nº OFF	REMARKS
	A	BEARING BLOCK	CAST IRON	4	
	B	" " COVER	" "	4	
●	C	COVER BOLTS	STEEL	8	
●	D	HOLDING DOWN "	"	24	
	E	OIL BOX COVER	"	4	

NOTE:– PARTS MARKED THUS ● MADE FROM BAR.
OTHER PARTS MACHINED WHERE MARKED *f*

DETAIL OF BEARING BLOCK.
SCALE:– ½" = 1, INCH.

DRAWING Nº 846/12.

CONTRACT Nº 846.

FIG. 4

Allowing a bearing load of 120 lb. per sq. in. of projected surface, we find the length of bearing $l = \frac{3 \times 2240}{6 \times 120} = 9{\cdot}33$ in., say, $9\frac{3}{8}$ in.

Peripheral speed $v = \frac{6 \times 3000}{230} = 78{\cdot}3$ ft.-sec.

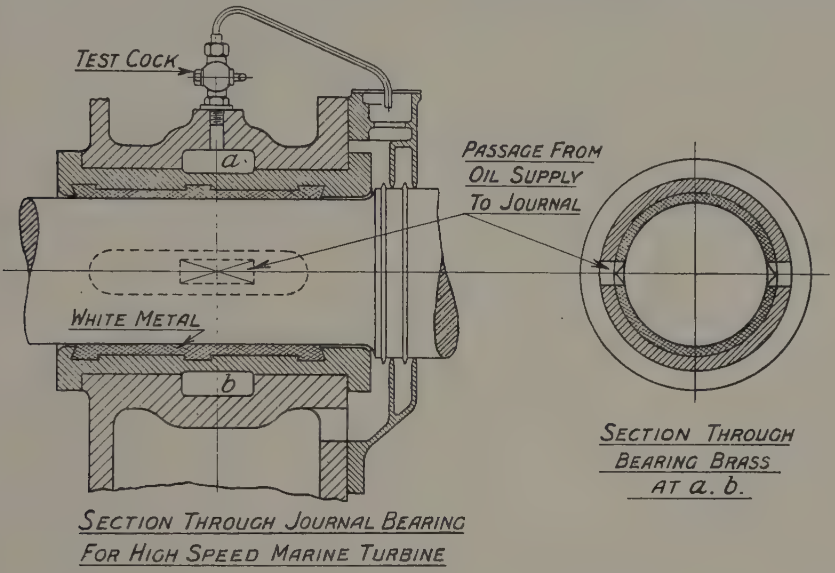


FIG. 5

Assuming American engine oil to be used at an average temperature of 110° F., then the coefficient of friction $\mu = {\cdot}028$.

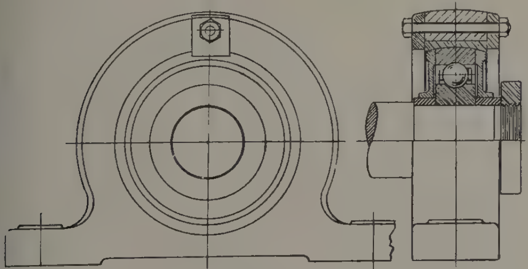


FIG. 6A. THE "HOFFMANN" BALL JOURNAL BEARING
(Hoffmann Manufacturing Co., Ltd., Chelmsford)

Heat generated in bearing

$$= \frac{\text{load} \times \mu \times v \times 3600}{778}$$

$$= \frac{3 \times 2240 \times {\cdot}028 \times 78{\cdot}3 \times 3600}{778}$$

$= 68{,}190$ B.Th.U.'s per hour,

then the heat to be radiated per sq. in. of projected bearing surface

$$= \frac{68{\cdot}190}{6 \times 9{\cdot}375} = 1213 \text{ B.Th.U.'s per hour.}$$

This quantity of heat is too great to be dissipated by radiation, and will require to be taken away either by a flow of cooling water in a jacketed block or, which is better, by feeding the bearing with a quantity of oil in excess of the amount required to maintain an oil film in the bearing. The latter being the better method of maintaining a cool bearing we may decide upon type 5 as the most suitable design for the purpose.

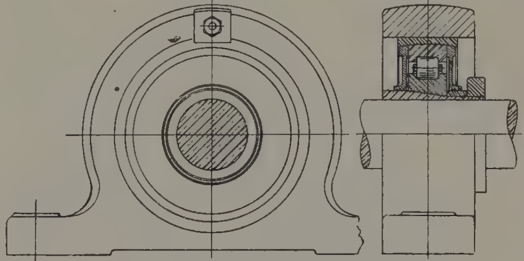


FIG. 6B. THE "HOFFMANN" ROLLER JOURNAL BEARING
(Hoffmann Manufacturing Co., Ltd., Chelmsford)

Draughtsman's Licence. The dimensioned drawing of the bearing block shown in Fig. 4 embodies one or two special features in the method of representing a machine detail. It will be observed that the end elevation shows the button, securing the white metal to the body, on the bottom of the bearing, whereas the side elevation gives a position of 30° to the vertical centre line. The correct position is as shown in the side elevation, and the incorrect projection to the end elevation has been resorted to for economy of line. To avoid any error in reading the drawing a note is added referring the reader to the side elevation for the correct vertical position of the button.

Further, it will be noticed that the side elevation is slightly more than half sectional. This extension of the sectional portion of the view has been adopted to provide a clearer view of that side of the oil box having the greater detail.

Another point to notice is the dimensioning of the position of the holding down bolts. The position of one bolt is given relative to one corner of the base, it being understood that the position of the bolts at the other corners will be similar.

EXERCISE

Make a finished drawing of the bearing block, Fig. 4, giving a half sectional side elevation, an outside end elevation, and a plan with the top half of the block removed.

FUEL

By Arthur Grounds, B.Sc., A.I.C., Assoc.M.I.Min.E., A.M.C.T.

LESSON VII

BOILER PLANT LOSSES

We have seen that probably the greatest loss which occurs on any boiler plant is that due to the escape of hot gases up the stack, carrying with them a large quantity of sensible heat, and we have also seen that the admission of too little air will lead to the formation of the gas, carbon monoxide, with a further consequent loss of available heat of the fuel. Heat is also lost in the ash which is discharged from the grates, and by the necessity of driving off any moisture in the fuel as burnt. Radiation losses can largely be eliminated by the judicious use of good lagging material on steam lines, over the crowns and drums of the boilers, and, in the form of mattrasses, on the economizers. The brickwork of the boiler setting should be examined carefully periodically for inleakage of air, using a frame of laths, covered with parchment, a candle being mounted behind a 2 in. hole cut in the centre of the parchment. Wherever leakage occurs, provided the plant is under draught, and not under pressure, the flame will be sucked in towards the brickwork, and in this way, a large expanse of brickwork can be run over quickly. Where such leakage is found to occur, the brickwork should be washed over with a mixture of hot tar and Portland cement. Where very good bricks have been used in the first instance, it is a good plan to wash over the brickwork periodically with a solution of silicate of soda, as this forms an impervious skin, preventing inleakage of air.

Other losses may occur in indirect ways. For example, although the actual conditions of combustion may be so carefully regulated that a figure of 12 per cent of carbon dioxide is being obtained in the flue gas, the efficiency of the plant may be low, due to the deposition of scale on the inside of the boiler, or to the formation of soot deposits on the outer shell of the boiler. In either case, a pyrometer inserted at the base of the stack would quickly show when such undesirable conditions were apparent, since the temperature of the exit gases would rise owing to the insulating nature of the deposits of scale or soot, the heat being prevented from passing freely through the shell of the boiler. The question of water softening is a very comprehensive one, and the reader must be referred to one of the standard works on the subject for detailed information. Sooty and tarry deposits should always be removed as often as practicable, since a layer of such an oily or tarry deposit of, say, $\frac{1}{8}$ in. thickness, is equivalent, as far as heat transmission goes, to slab of cast iron of about 4 to 5 in. in thick-ess. The sooting up of boiler tubes is particu-arly likely to give trouble in water-tube boilers, ut the danger is present even in Lancashire, ornish, or marine boilers. In almost all oiler plants, except the very smallest using, ay, a very small vertical boiler, it pays to install *nd use* a carbon dioxide indicator or recorder, nd a pyrometer, since the information which hey give is so valuable if interpreted correctly. Iost carbon dioxide recorders of the older type epend on the fact that carbon dioxide combines vith caustic potash to form potassium carbon-te. A measured volume of the gas is, there-ore, passed into a chamber containing caustic otash solution, the carbon dioxide dissolves in he solution, and the remaining gas is then assed back into the measuring vessel where it s measured at the same pressure as was the riginal gas. The diminution in volume gives he volume of carbon dioxide removed. In arbon dioxide recorders, the residual gas is nade to operate a piston which is coupled up to pen, the latter then marking on a chart the ravel of the piston, this being marked off in 'Per cent CO_2" instead of inches. In the lectrical carbon dioxide recorders which are ow coming into use in large numbers, the lifference in thermal conductivity between arbon dioxide and air is made use of, two dentical platinum spirals losing heat to the valls of their containing cells which contain, on he one hand, air, saturated with moisture, and n the other hand, the flue gas under test. The esistance of the spirals varies with the tempera-ure, and the two spirals form two arms of a Vheatstone's bridge system. Full particulars f the mode of operation of these instruments an be obtained from the makers, e.g. The Cambridge Instrument Co., Ltd., Walker, Crosweller & Co., Ltd., Siemens Schuckert, Ltd., James Gordon & Co., Ltd., The Sarco Trading Corporation, Ltd., etc. The temperature of the exit gases from the plant may be measured by means of a strong mercury thermometer, armoured to withstand rough usage, and reading up to, say, 1,000° F., or by means of a base metal thermo-couple, connected in circuit with a millivoltmeter, calibrated direct in degrees centigrade.

Care should be taken to see that all instruments are so mounted that they give figures representative of the average performance of the plant. For example, in installing gas sampling pipes in horizontal flues, there is a tendency for the hotter gases to ascend to the crown of the flue, and if the pipe be mounted low down in the flue, the samples of gas withdrawn through such a pipe may show remarkable consistency, owing to the fact that "dead" gas is being withdrawn for the tests. In a vertical flue, this danger is not likely to occur. Samples should not be taken near any sudden bend, as here again pockets of dead gas are liable to be found. Similarly instruments should not be mounted near dampers for the same reason. Quite recently, an instrument has been devised which shows the radiation from any desired surface, giving readings direct in British Thermal Units lost per square foot per hour. The fuel supply is usually weighed direct, as in the case of coal, or is measured in volume, as with gas and oil, whilst the water supply to the boilers can be either weighed direct in tanks or can be measured by means of one of the reliable meters, of which there are a large number on the market at the present time.

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON V

ECONOMIZERS

The function of an economizer is to impart a certain amount of heat to the boiler feed water before it actually enters the boiler.

It is placed in the path of the furnace gases after they have left the boiler on their way to the chimney.

For the smaller installations the economizer is generally common to all the boilers, i.e. the furnace gases of all the boilers pass into a common flue in which the economizer is installed.

For larger installations it is now usual to arrange for a separate economizer for each boiler, and for this to form an integral part of the complete steam raising unit.

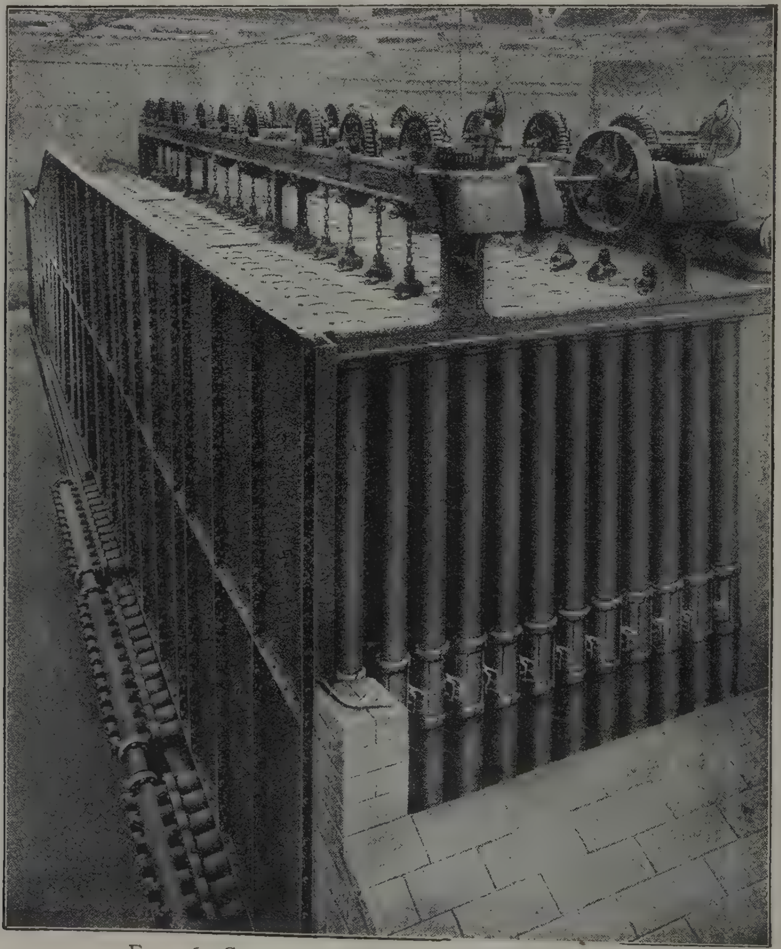


Fig. 26. General View of Green's Economizer
Showing scraper gear

With cylindrical boilers the temperature of the furnace gases as they leave the boiler is probably from 600 to 700° F., and after passing through an economizer this temperature may be reduced to about 400° F., showing that a very considerable proportion of heat in the ue gases, which would otherwise be wasted, can e usefully employed by this means.

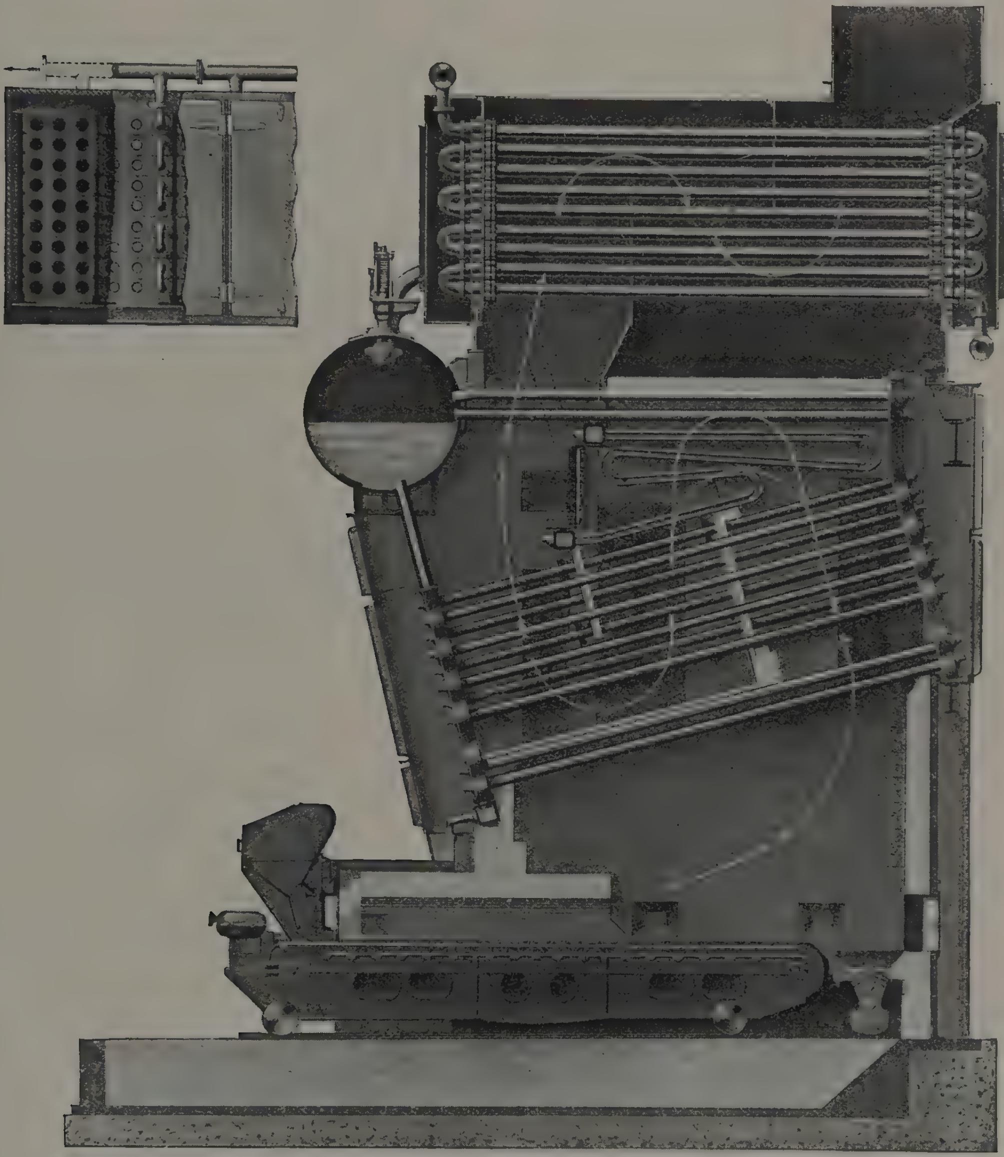


FIG. 27. GREEN'S PATENT TRI-TUBE HORIZONTAL ECONOMIZER
Installed over a Babcock & Wilcox C.T.M. boiler

Even on a well-designed water-tube boiler, here the temperature of the gases leaving the boiler is probably not more than 500 to 550° F., there is usually still an appreciable advantage to be gained by the installation of an economizer.

It is not desirable that feed water should be introduced into an economizer at a lower temperature than 100° F. Where condensate from steam turbines or engines is available, and steam traps, drains, pump exhausts and other auxiliaries are made use of for feed heating, it is often found possible to obtain an initial temperature of about 120° F., and this should be aimed for whenever possible.

The disadvantages of using a very cold "feed" are that all furnace gases contain a certain amount of moisture, and this, striking against the cold surface, condenses on the tubes, and being invariably slightly acid sets up external corrosion and wastage.

There are several ingenious methods of mitigating this trouble. One plan is to put the first few rows of tubes in "series," so that the water is only cold in the first row or two and any wastage of the tubes is definitely confined to these sections.

Another plan is to "short-circuit" some of the heated water from the hot end of the economizer and allow it to mix with the cold water as it enters, and by this means a more even temperature over the whole apparatus is ensured.

In the early days of economizers the tubes were always constructed of cast iron, but in the larger installations of to-day it is common practice for steel tubes to be used. This particularly applies to plants working at the higher pressures, as although cast iron has proved quite suitable for pressures of up to, say, 250 or even 275 lb. per sq. in., there are obvious difficulties when pressures above this are being considered.

For superposed economizers, which are practically standard practice on the larger modern steam-raising units, the question of weight is also very important, and in this respect the steel tube shows to advantage.

The thinner steel tube has, however, the disadvantage of being more quickly perforated should internal corrosion, due to the presence of oxygen in the water or other causes, be set up, but the real remedy for corrosion is obviously to remove the cause rather than by relying on a thicker tube to withstand corrosion a little longer than a thinner one.

Green's Economizer. The standard Green's economizer is made up of $4\frac{9}{16}$ in. external dia. cast-iron vertical tubes—usually 9 ft. long, although for special conditions they may be longer—pressed into cast-iron headers and forming sections of various widths.

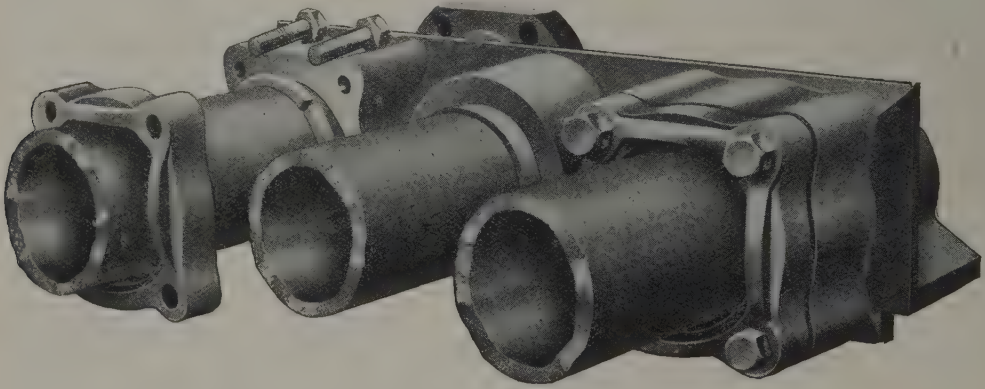


FIG. 28. GREEN'S TRI-TUBE ECONOMIZER HEADER
Showing ringstays

The tubes are turned slightly tapered at either end to fit corresponding borings in the headers.

Secure metal-to-metal joints are effected by hydraulically pressing together a complete section at both ends simultaneously.

Sections are erected vertically side by side to any desired length, each section having a connecting branch at top and bottom, on opposite sides to which connecting pipes are attached by flanged and bolted joints.

Access is obtained to each tube through hand holes in the upper headers, these being tapered from the inside to receive similarly tapered lids.

The lids are drawn into position by a special lifting key and the pressure of the water maintains a perfectly tight joint. A smart tap with a hammer from the outside will knock in the lid when it is necessary to obtain access to the tubes for cleaning purposes.

In each section there is a master lid slightly arger than the others to facilitate their entry or removal. The cold water is admitted at the bottom of the economizer nearest to the outlet of the furnace gases, so that the colder flue gas meets the colder water and vice versa.

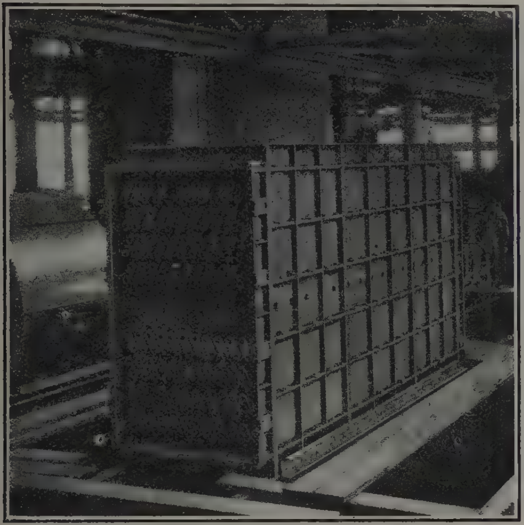


FIG. 29. FOSTER ECONOMIZER
Showing arrangement of tube end connections

Very special arrangements are made for the removal of soot from the outside of the economizer tubes by scrapers, which entirely surround each tube and continuously travel from top to bottom. This movement is accomplished by supporting the scrapers on adjacent rows by means of carriers and guards. These are bolted together and suspended in pairs by a chain which passes up between the sections and over a chain wheel. The rotation of these chain wheels by suitably geared and reversible mechanism causes one set of scrapers to travel upwards while the other is travelling downwards, and at the end of each travel the process is reversed.

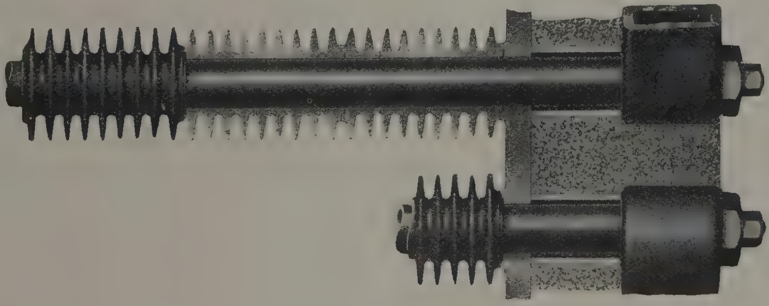


FIG. 30. FOSTER ECONOMIZER
Details of tubes

To meet the demand for economizers suitable for the higher pressures, Messrs. Green have introduced a ringstay tube having a shoulder turned at the end of the taper. By means of a split ring and plate and four set-screws, anchorage is secured to the headers, which are specially stiffened for the purpose, and it is claimed that tube joints thus made are 100 per cent stronger than the ordinary pressed and tapered joint.

This ringstay joint was more particularly designed for use with Messrs. Green's patent tri-tube horizontal economizer, for use in conjunction with high-duty self-contained steam-raising units.

In this design each section consists of three tubes only, the sections lying horizontally and being connected at their ends by short connecting pieces, either in series, parallel, or a combination of both. The smaller section is especially convenient for superposed economizers where the handling of heavy weights is more difficult.

The Babcock and Wilcox Economizer. This apparatus is composed of steel tubes connected at either end into steel headers or uptakes.

The tubes are usually arranged with a slight inclination from the horizontal, but can also be arranged vertically.

As in the case of the Babcock and Wilcox boiler, the economizer headers are "staggered" in such a way that the tubes in alternate rows are symmetrically opposite the space between tubes in the adjacent rows.

This economizer is almost always superposed directly above the boiler, and forms an integral portion of the complete steam-raising unit.

Access to each tube for cleaning purposes is obtained through hand holes in the headers, and exterior cleaning is performed by soot blowers or steam lances.

The economizer tubes are 4 in. in diameter, and of approximately the same length as the boiler tubes.

The standard arrangement allows for two sets of baffle plates, so that the furnace gases make three passes through the tube sections.

Messrs. Babcock & Wilcox have also recently introduced a high duty economizer designed on the contra-flow principle. In this apparatus the tubes are also of steel, but are reduced to 2 in. diameter, with a consequent higher water velocity.

The arrangement makes for compactness, and has been adopted in many instances where air heaters are also included in the installation.

The Foster Economizer. In the Foster economizer, specially gilled tubes have been adopted in order to obtain a maximum heat transference. The actual tubes are of solid drawn steel of small diameter with cast-iron rings shrunk on the outside, the rings being spigoted to completely protect the tube over its entire length.

The tubes are placed horizontally, and expanded at either end into steel headers and manifolds. Cast steel connecting boxes are connected at either end of the rows in such a way that the water travels through the tubes in series at high velocity, and in the opposite direction to the flow of the flue gases.

The tubes are slightly "staggered" vertically and are individually accessible through hand holes, the joints being made by means of copper gaskets, and a tapered steel plug.

For cleaning the exterior of the tubes, steam-operated soot blowers are provided.

The complete economizer is enclosed in a steel casing, this being made up of double sectional plates set about 3 in. apart, and the intervening space being fitted with insulating material.

Both flue gas and feed water pass through the apparatus at a high velocity, and a maximum efficiency is claimed.

The internal resistance or draught loss, in these high duty economizers is naturally higher than where larger tubes with greater spaces between them are employed, and this should be allowed for when determining the amount of chimney draught necessary for the efficient operation of any particular installation.

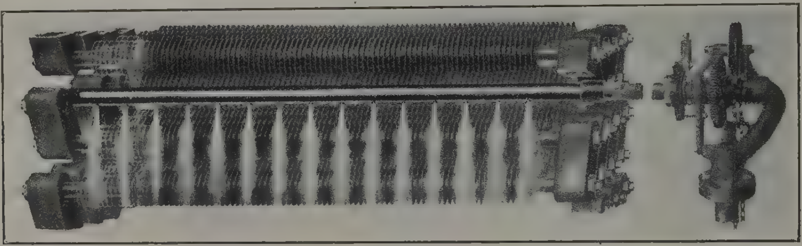


FIG. 31. FOSTER ECONOMIZER

Sectional view showing rotary steam soot blowers in action

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.Inst.N.A.

LESSON III

CLASSIFICATION OF STEAM TURBINE TYPES—(*contd.*)

The Velocity-compounded Turbine. Suppose we arrange purposely that the ratio of blade velocity to steam velocity is small. Then the velocity diagram, Fig. 8, shows that the final absolute velocity CE from a single-row impulse turbine is not only high in magnitude, but is directed backward at a relatively small angle to the plane of the wheel.

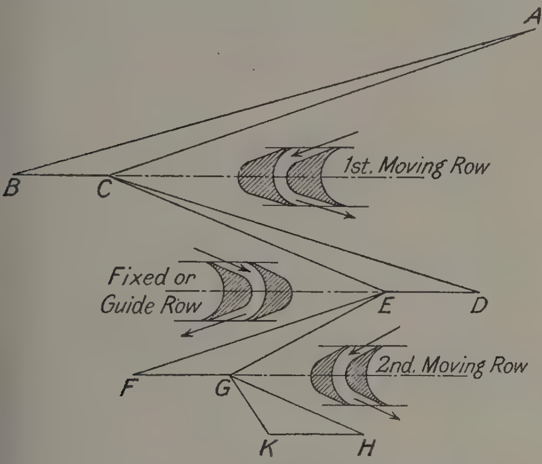


Fig. 8

If we now arrange a suitable number of guide blades close to the afore-mentioned moving blades, as shown in Fig. 9, these may be caused to redirect the jets of steam, which issue from the moving blades, on to a second moving row of blades wherein much of the remaining kinetic energy $\left(\frac{CE^2}{2g}\text{ ft.-lb. per lb.}\right)$ may be absorbed in performing useful work on the turbine shaft. This is the principle of *velocity-compounding*.

The changes of velocity are shown clearly in Fig. 8. It will be noticed that the inlet and outlet angles of the blades gradually increase as the velocities diminish. Fig. 8 is drawn for two moving rows of blades. Rotors having two rows of moving blades are referred to as two-row wheels. Fig. 9, on the other hand, is a diagrammatic sketch of a three-row wheel. It should be observed that in the true impulse turbine, the steam pressure is constant throughout the blade system, except such local changes as must necessarily occur in the blade channels.

Pressure-velocity-compounded Turbine. The velocity-compounded turbine just described above may also be compounded for pressure in stages. We then have two or more wheels, each fitted with two or more complete rings of moving blades and with an appropriate number of guide or redirecting blades arranged between them and attached to the fixed casing of the machine. The wheels are keyed to the same shaft, and the various stages are arranged in series. The type is represented diagrammatically in Fig. 10.

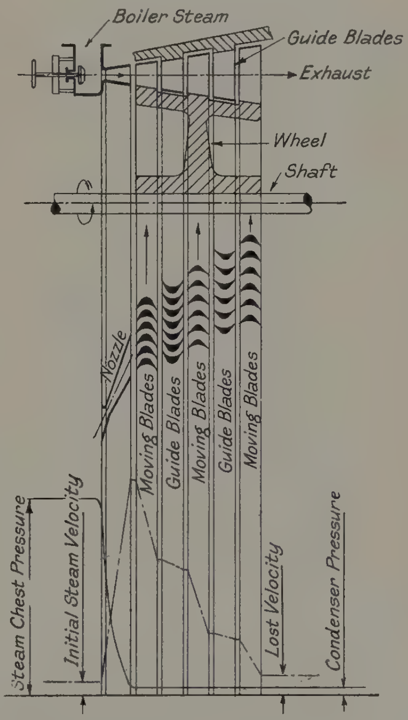


Fig. 9. Diagrammatic Arrangement of Velocity-compounded Impulse Turbine

Steam is admitted to the steam chest, and in flowing through the nozzles its pressure falls and its velocity increases. As it passes through the first stage blades, its pressure remains constant and its velocity diminishes, as work is being done on the blades at the expense of the kinetic energy of the steam. In passing through the second stage nozzles, the steam again suffers a pressure drop and its velocity increases. The constancy of pressure and gradual reduction of velocity which characterizes the first stage are also true of the second and succeeding stages.

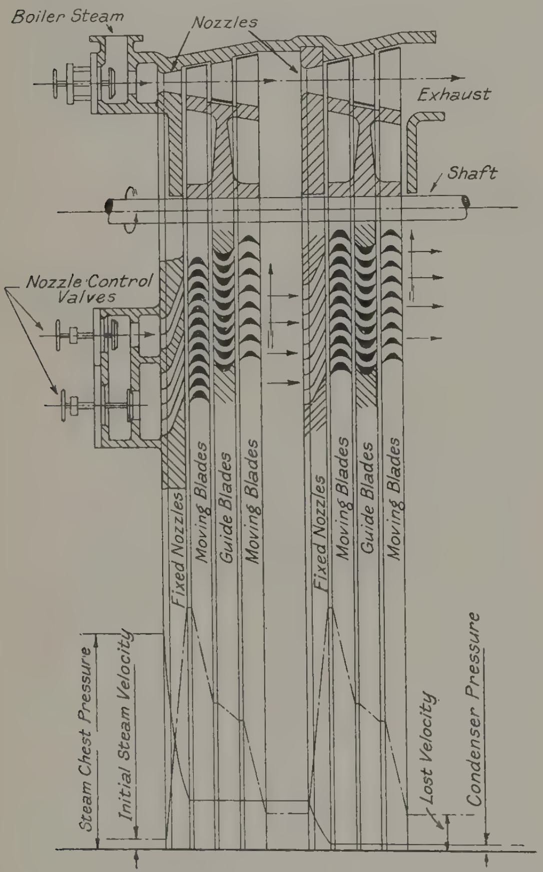


FIG. 10. DIAGRAMMATIC ARRANGEMENT OF PRESSURE-VELOCITY-COMPOUNDED IMPULSE TURBINE

It may be mentioned here that the main advantage of velocity-compounding is the large "heat drop" which it permits for a given blade velocity. Consider the velocity diagram shown in Fig. 11 for a two-row wheel. In drawing this velocity diagram, we have made two assumptions, viz.: (1) that no losses of energy occur within the blade channels, and therefore that the relative velocity at outlet from the blades is equal to the relative inlet velocity; (2) that the inlet and outlet angles of any one row of blades are equal.

Let CE^1, E^1G^1, and G^1K^1 be made equal to the blade velocity. Then in the triangles ACE^1, CDE, because $AC = CD$, $\angle ACE^1 = \angle E^1CD = \angle CDE$ and because $CE^1 = DE$, the triangles are congruent. In like manner, it may be shown that the pairs of triangles AE^1G^1, EFG; AG^1K^1, GHK; are respectively congruent. That is, $AE^1 = CE$;] $AG^1 = EG$; and $AK^1 = GK$.

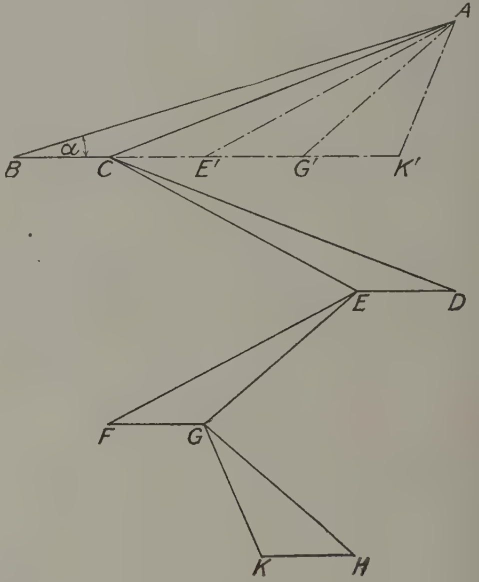


FIG. 11

From the well-known principle of the conservation of energy, we may write down the following statement—

Provided there is no change of pressure of the steam,

Initial kinetic energy

= (final kinetic energy) + (losses) + (work done).

i.e. work done

= (initial kinetic energy—final kinetic energy)—losses.

In this instance, we have assumed no losses in the blades.

∴ work done per lb. of steam

$$= \frac{AB^2}{2g} - \frac{GK^2}{2g}$$

$$= \frac{AB^2}{2g} - \frac{(AK^1)^2}{2g}$$

Clearly, the work done, and hence the efficiency, will be a maximum when AK^1 is a minimum, i.e. when AK^1 is perpendicular to BK^1. This condition is only fulfilled when BK^1 which s equal to $4u$ equals $V_1 \cos \alpha$; in other words, he efficiency is a maximum when

$$\rho = \frac{u}{V_1} = \frac{\cos \alpha}{4}$$

By a similar argument for a three-row wheel t may be shown that the maximum efficiency s obtained when $\rho = \frac{\cos \alpha}{6}$ and generally, for u rows of *moving* blades, when $\rho = \frac{\cos \alpha}{2\mu}$.

If, now, we compare a simple impulse and a wo-row impulse turbine having *the same blade elocity*, we find that if each is to have the maximum efficiency of its type, the steam elocity in the two-row turbine will be twice hat in the simple impulse machine and, as nergy varies as the square of the velocity, he kinetic energy utilized is just four times that which can be utilized in the single-row wheel. Thus, if two multi-stage turbines, one equipped with single-row wheels and the other with two-ow wheels, compounded for pressure in each ase, are to operate between the same limits of ressure, then the number of single-row stages equired will be four times the number of two-ow stages.

The pressure-velocity-compounded turbine vas invented by the American engineer, Ir. C. G. Curtis, and became a standard power tation machine for a number of years. Its fficiency, however, is lower than that of the ulti-stage one-row impulse turbine, and large achines of this type are never built nowadays.

In order to maintain reasonably small outlet ngles and so obtain an economical velocity hange in the moving blades, the blade heights re progressively increased, as shown in the iagrammatic sections in Figs. 9 and 10.

EXAMPLE. The following particulars refer to a wo-row velocity-compounded impulse turbine—

Initial steam velocity, 1,500 ft. per second.
Mean blade velocity, 350 ft. per second.
Nozzle angle, 20°.
Outlet angle of first row of moving blades, 22°.
Outlet angle of guide blades, 28°.
Outlet angle of second row of moving blades, 40°.

$$k = \frac{\text{relative outlet velocity}}{\text{relative inlet velocity}} = 0{\cdot}85 \text{ for all}$$

blades.

Draw the velocity triangle and calculate the work done in each row of moving blades per lb. of steam. Calculate also the efficiency of the system.

The velocity triangle is drawn in the usual manner and is shown in Fig. 12. The inlet angles of the various blades are also shown in this diagram.

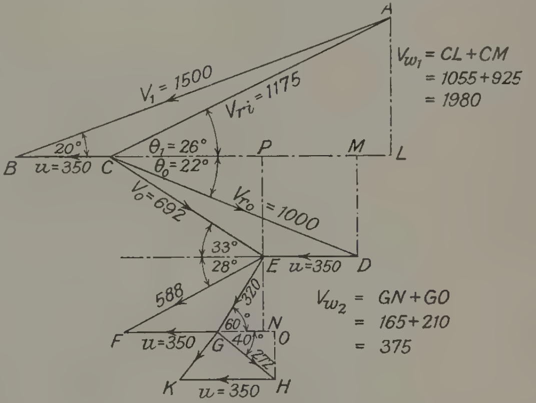


FIG. 12

For the first row of moving blades,

$$V_{w1} = BL + CP$$
$$= CL + CM \text{ (since } BC = PM\text{)}.$$

Hence, $V_{w1} = 1055 + 925 = 1980$ ft. per second, and thrust on blades *per lb. of steam per second* will equal $\frac{V_{w1}}{g} = \frac{1980}{32{\cdot}2} = 61{\cdot}5$ lb. weight.

∴ work done $= 61{\cdot}5 \times 350 = 21{,}500$ ft.-lb.

For the second row of moving blades, the corresponding figures are 375 ft. per second, 11·65 lb. weight, and 4070 ft.-lb. of work.

∴ Gross work done on blades

$$= 21{,}500 + 4070$$
$$= 25{,}570 \text{ ft.-lb.}$$

$$\text{Initial kinetic energy} = \frac{1500^2}{64{\cdot}4} = 35{,}000 \text{ ft.-lb.}$$

$$\therefore \text{ efficiency} = \frac{25{,}570}{35{,}000} = \underline{73 \text{ per cent.}}$$

RECIPROCATING STEAM-ENGINES

By Engineer Lieut.-Commander T. Allen, R.N. (s.r.)

LESSON VII

CHOICE OF PRIME MOVERS

In certain cases local conditions, such as space available, lack of condensing facilities, available fuel, remoteness of facilities for repair, etc., may limit the choice of a prime mover to one or other of the types which have been discussed.

In general, however, the determining factors are: efficiency, reliability, cost of running and maintenance, capital cost and space occupied; the order of these factors varying in accordance with individual requirements.

Where appreciable quantities of process or heating steam are required, the opportunity occurs to avoid the whole or part of the serious loss represented in condensing engines by the rejection of heat to the condensing water. In these cases, the question of economy is of paramount importance, and back pressure or extraction engines should be installed. For condensing engines, economy of steam is improved by separation of the valves, effective governing, reduction of clearance spaces, and avoidance of excessive cylinder temperature variation. These qualities are possessed in varying degree by Unaflow engines, drop and Corliss valve engines with compounded cylinders, and good high speed engines. These engines will give a high comparative thermal efficiency when supplied with superheated steam and installed under satisfactory working conditions. Their reliability, low maintenance and depreciation charges have been proved in practice and, for general industrial purposes, these types are particularly applicable. The

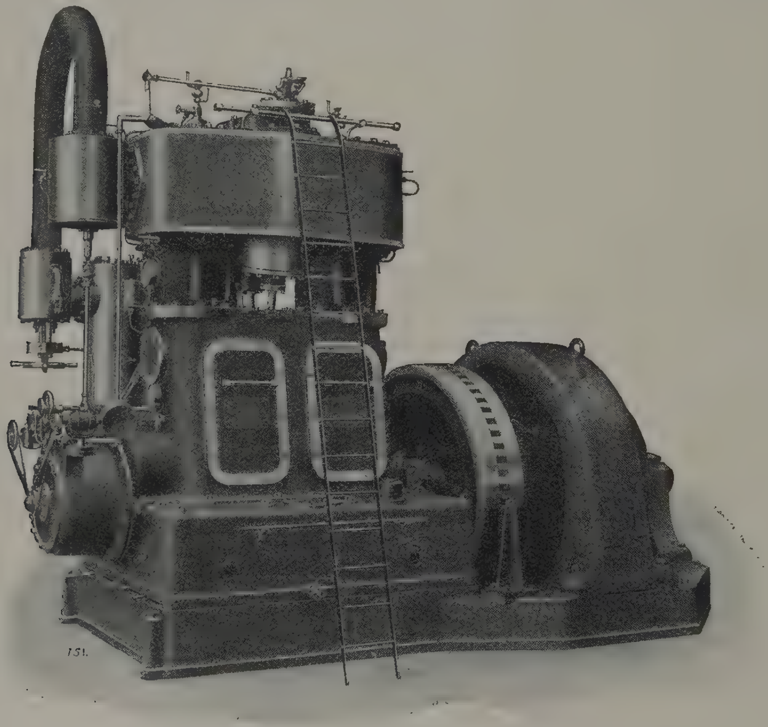


Fig. 22. "Belliss" Patent Two-crank "V" Type Compound Inclined Valve Self-lubricating Engine

team consumption of the Unaflow engine is omewhat lower than that of the compound ngine, and it possesses the further advantages f simplicity and small space requirements. It s, therefore, being installed in an increasing umber of cases to the exclusion of the older ypes. Experience also shows that the Corliss ngine has been practically superseded by the rop valve engine. The high speed engine is sed for a great variety of generating, ventila- ng, pumping, and factory driving purposes, nd has the advantages of small space require- ents and low initial cost, owing to the smaller eights and sizes made possible by the increased evolutions.

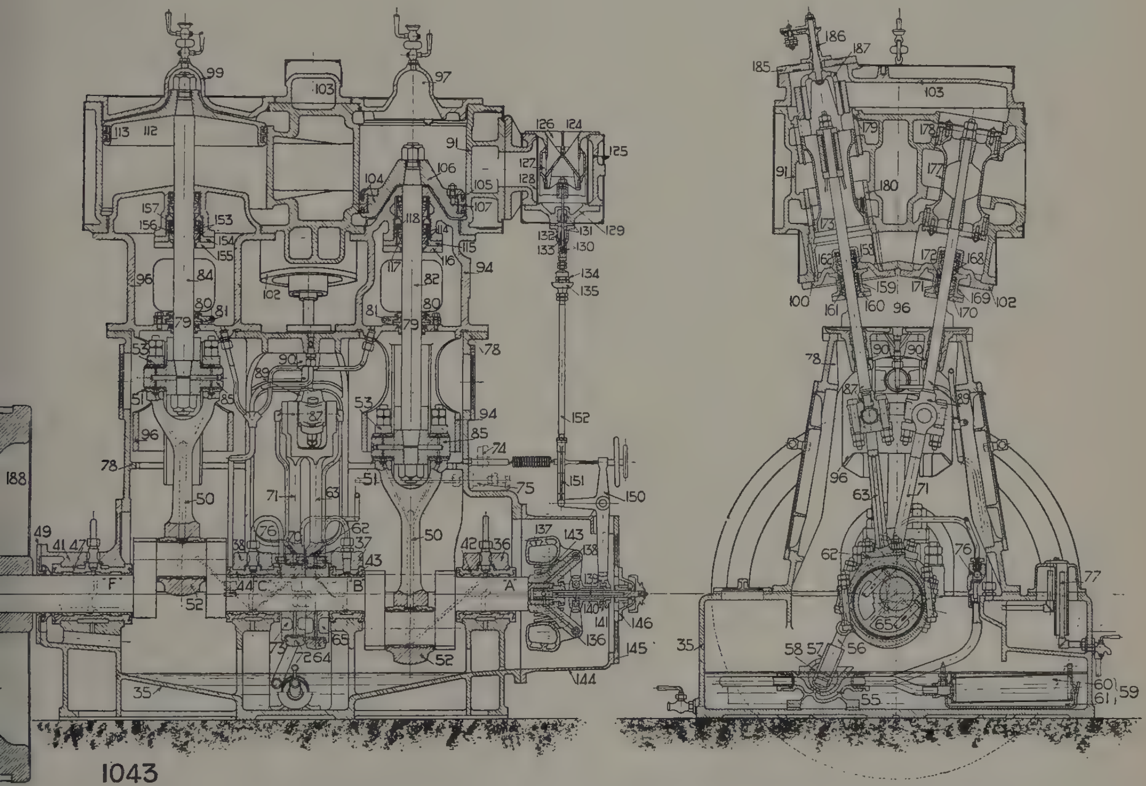


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Fig. 23. Section of "Bellis" Patent Two-crank "V" Type Compound Inclined Valve Self-lubricating Engine

106.—H.P. piston.	173.—H.P. valve.	127.—Throttle valve.	58.—Forced lubrication pump.
112.—L.P. piston.	177.—L.P. valve.	143.—Governor.	48.—Crankshaft.

A good example of this type is shown in igs. 22 and 23, which give outside and sectional ews respectively of a two-crank "V" type mpound engine made by Messrs. Belliss & orcom, to whom the writer is indebted for the illustrations. The design as shown avoids the necessity for outside receiver pipes as the high pressure valve cuts off on inside edges and exhausts directly to the low pressure valve. Both valves are of the piston-slide type, driven by separate eccentrics and rods at an angle on either side of the engine centre line, and accommodated in separate valve chests between the cylinders. Normally, the cranks are set opposite to each other, the steam being admitted simultaneously to the top of one cylinder and the bottom of the other, so as partially to balance the reciprocating masses and reduce the stress on the bearings, and the tendency to vibration. If desired, however, the cranks may be at right angles. A centrifugal governor with spring drive and hand speed adjusting device is fitted on the crankshaft, and works in conjunction with a throttle valve and an automatic expansion gear so as to give throttle governing at light loads, automatic expansion at heavy loads, and a combined effect at intermediate loads. The main bearings, crank and crosshead pins and guides, eccentric and valve pins, and the working parts of the governor are forced lubricated through a system of passages and pipes, receiving strained oil under pressure from a valveless pump immersed in oil in the crank pit.

Application of Steam-engines to Industry. Each engine type is more particularly applicable to some special industrial duty, although the respective spheres overlap to a considerable extent.

For cotton and other textile mills, collieries, rolling mills, steel and copper works, generating stations of moderate power, flour mills and factories in which there is not a great demand for low pressure, heating or process steam, Unaflow and compounded condensing engines are suitable.

Pumping stations may have condensing Unaflow, compound, or triple expansion engines.

The steam extraction engine is specially suited to paper mills, bleach and dye works, woolcombing and weaving sheds, tanneries, and all industries in which power and varying quantities of low pressure process or heating steam are required.

Chemical and nickel works, laundries, public institutions, and similar undertakings requiring power and large quantities of low pressure steam may also find the steam extraction engine a suitable prime mover, although in these cases the back pressure engine often shows to better advantage. Steam extraction and back pressure engines will be dealt with in greater detail, in a separate lesson.

Transmission of Power. Power may be transmitted from steam-engines in a variety of ways, as follows—

Direct drive to generators, pumps, fans, or shafting from half couplings at one or both ends of the engine shaft, or by mounting alternators directly on the crank shaft alongside the fly-wheel.

Rope drive from the engine rope fly-wheel or —where the speeds are not suitable, e.g. Unaflow cold roll drive—from a rope pulley mounted alongside the fly-wheel or upon a shaft direct-coupled to the engine.

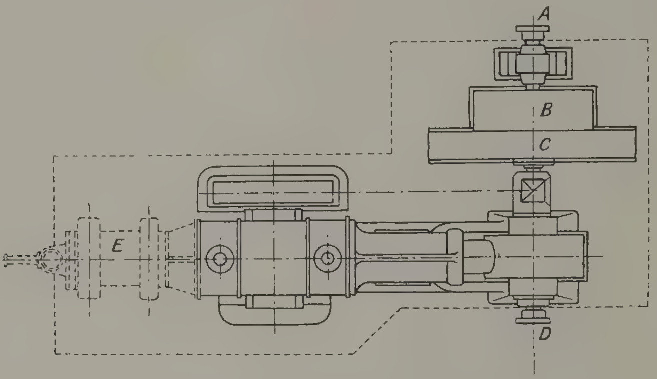

Fig. 24. Diagrammatic Outline Illustrating Engine Drives

Direct Drive: Through coupling *A*, or to alternator at *B*, with exciter at *A*.
Rope Drive: From rope fly-wheel at *C*, or rope pulley at *B*, alongside plain fly-wheel at *C*.
Belt Drive: From plain fly-wheel at *C*, or belt pulley at *B*, alongside plain fly-wheel.
Combined Drive: Through couplings *A* and *D*, with rope or belt drive from *C* or *B*.
Drive Through Tail Rod: Direct to pump or air cylinder at *E*.

Belt drive from the plain fly-wheel of the engine, or from belt pulleys arranged as for rope pulleys.

Gear drive from pinion or bevel wheel mounted directly on the crankshaft, or connected thereto by a coupling and shaft.

Combined drives, e.g. direct-driven generator at one end of crankshaft, direct-coupled shafting at the other end, and rope or belt drive from fly-wheel or from pulley mounted alongside.

Drive through piston-rod tail extension. Notably to hydraulic and pumping machinery, and to *air-blower cylinders* and compressors. Fig. 24 shows a diagrammatic outline of the various drives, and actual examples are illustrated in the photographs reproduced in the lessons.

Where direct drive is practicable, transmission losses are reduced to a minimum, but the engine speed limits the speed at which the driven element may run. This may result in a igher capital cost for the generator, pump, r fan, but this is offset to a great extent by ne cost and maintenance of the ropes, belts, earings and pulleys required for indirect drive, nd by the transmission losses. Direct drive ives a neater arrangement and requires less pace. Where the engine is directly coupled to ain shafting or generators, special precautions hould be taken to secure exact alignment.

Where gear drive is adopted, e.g. speed reducng or increasing sets to rolls, pumps, fans, enerators, etc., the gears should be suitably nclosed and lubrication may be effected by an xtension of the engine forced lubrication ystem—if such is fitted. If the interiors of rge gear cases are subjected to a slight vacuum —by means of small vacuum pumps—leakage of il through the joints may be eliminated.

It should be noted that where alternators are riven direct from the crankshaft, the speed of he engine is fixed by the periodicity of the urrent, and the engine can, therefore, only be un at certain fixed speeds.

Indirect drive through the medium of steel elts has been adopted successfully in a number f cases. For rope drives, round cotton ropes re the usual standard, but under roped drives nay often be improved by using ropes of quare section.

Arrangement of Plant. The layout of steamngine installations should be such as to give mple access to all parts, both in the engineoom and in the basement. A hand-operated verhead crane facilitates erection and overhaul. oundation bolts are best arranged to be apable of insertion or removal after the engine erected, and suitable handholes should be rovided in the base of the foundations to xpose the securing nuts and plates. All parts f the engine should be capable of easy disnantling for examination or repair, and the esign of all castings subject to high temperaure should be such as to reduce distortion. Ieat losses are reduced by covering the cyliners and steam pipes with magnesia or other on-conducting composition of sufficient thickess, reinforced and covered where necessary, nd suitable for the highest temperatures likely be experienced. Some engineers prefer the des of fly-wheels to be enclosed in planished eel laggings, and these have the advantage of educing "windage" effects.

A steam receiver with adequate drainage rrangements should be inserted in the steam ipe close to the engine, and where surface condensers are used, or the discharge of jet condensers is required for boiler feeding purposes, oil separators or filters are required in the exhaust or feed pipes. If injection water flows to jet condensers from some higher level than the engine, ample vacuum breaking arrangements are necessary to prevent mishaps due to flooding of the cylinder. It is best, where possible, to take the water from a lower level, but where the source of supply is elevated, the pressure may be broken by arranging a special injection well in the basement, with a float-controlled inlet and an overflow to a natural drain. Trap discharge and drainage sumps in the basement, should also have a flow by gravity to the drain.

Foundations. Foundations are, nowadays, usually constructed in a concrete mixture of Portland cement, sand and rubble.

The design should provide for a rigid structure built upon a concrete raft of a thickness corresponding to the nature of the subsoil. Adequate openings are required for access to all parts requiring examination or adjustment, but undercutting and weakening of the sections should be avoided as much as possible, especially underneath main bearings and frames. Where generators are direct-coupled to the crankshaft, it is advisable to construct the generator foundation in one with the engine foundations. Similarly, no cleavage should occur where provision is required for fly-wheel pits and outer bearings. The finished height of the foundation block may be a little less than the final level, so that during erection, the engine may be placed in alignment above the foundation, and the space filled with a ramming of Portland cement and granite chippings.

Erection and Running. Provision is made by means of sliding cylinder feet for the movement of the engine when heated and, during erection, this movement should be borne in mind when setting the position of eccentrics and rods likely to be affected. Eerction should only be undertaken in a reasonably clean and enclosed engine house, so as to reduce the access of dirt and foreign matter to the working parts. Upon first starting up, the lubrication system should be thoroughly flushed through and full speed attained slowly. During ordinary service, careful attention is necessary to warming-up, drainage, lubrication, governing, and valve setting, and defects should not be allowed to accumulate.

Testing. Engine tests should be made with properly calibrated instruments in perfect working order. In particular, the indicators should be examined, oiled, and certified free and steam-tight. The indicator drive also requires attention to see that inaccuracies in the cards are not caused by a faulty drive. In the case of surface condensing engines, the steam consumption may be obtained by weighing the condensate, but with jet condensers it is necessary to measure the water fed to the boilers. Special boilers should be isolated during the test for the sole purpose of supplying steam to the engine. All joints in the steam pipes should be steam-tight, and branches in the pipes blanked off. The water level in the boilers should be maintained as uniform as possible, and the level at the commencement and end of the test should be the same. Steam trap and drain discharges should be collected and weighed where necessary, and the engine loads, steam pressures, and temperatures maintained as uniform as possible. The chief readings required are: steam pressures and temperatures, condenser cooling water inlet and outlet temperatures, vacuum, barometric pressure, engine speed, and condensate or feed water quantities. Indicator diagrams should be taken at regular intervals, and, as far as possible, all observations should be taken simultaneously. Where the engine drives an electric generator or alternator, readings of the ampere, volt, power factor, and unit meters may also be recorded as necessary. During the ordinary service of the engine, a selection of the readings may be taken at intervals, and recorded in the engine house log.

WORKS ORGANIZATION AND MANAGEMENT

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LESSON VII

THE PLANNING DEPARTMENT

Relation to Other Departments. It is not always considered necessary to organize a separate department to exercise the function of "planning," and instead, the duties of this department are often therefore combined with those of some other department, such as the production department. The fact that it may be combined with another department does not alter the nature of its activities, however, and it will be clearer to study these, assuming that the department is organized on a separate basis. The place of the planning department in the general organization may be seen from the following considerations—

1. The order department is responsible for the receipt and issue of customers' orders to be manufactured.

2. The design and drawing department determines the form of the article to be manufactured.

3. The planning department arranges the sequence of operations, and the details of how the article is to be made.

4. The rate-fixing department decides how long an operation should take and, consequently, how much should be paid for it.

5. The production department regulates the order in which work has to be done according to its urgency.

Necessity of Planning. Planning of a sort has always been a necessity, of course, the difference being that the introduction of a planning department means the substitution of order and efficiency for uneconomical methods. Lack of proper planning means the increase of the following losses—

1. Machine tools standing idle whilst the operator, the foreman, and possibly other persons, discuss the ways in which a new piece of work may be done.

2. Tools, etc., will not be prepared in advance.

3. The best method will probably not be used.

4. The methods decided upon are not recorded and, therefore, have to be worked out again when the order is repeated.

Location of Planning Department. If possible, he planning department should be located lose to the drawing office, as it is desirable that here should be close co-operation between these wo departments. Every new drawing must ass through the planning department in order hat the best methods of manufacture may be etermined. If there is close co-operation many f the planning difficulties may be removed by light alterations in the design, and the designer lso gains by being in touch with the actual nanufacturing requirements. It will be noticed hat the planning department does not come into he "line" organization, but exists as an dvisory staff, which makes standards for new vork and has nothing to do with repetition rders.

Records Required. The work of the planning epartment will be made much easier if good ecords are kept such as the following—

1. *Machine Tools.* These will be classified ccording to their type and the work for which hey are best suited. Full particulars of each nachine must be given so that the planner will e assisted in making the right choice of the nost suitable tool for each operation. Owing o there not being a sufficient number of cer-ain tools available it may be necessary for the lanner to specify alternative tools, but the *ight* tool should always be specified as first hoice in order to draw attention to any out f balance in the numbers of each type of tool. uch out of balance can then be rectified when ools are next purchased.

2. *Equipment.* The record of the equipment as important as the record of the tool itself, s it will completely modify the capacity and ccuracy of the machine.

3. *Small Tools.* The record should include oth standard and special tools, its use being o promote standardization.

4. *Component Parts.* In this case the same ecord as that used by the drawing office may e used—and for the same purposes.

5. *Processes.* The work of the planner will e greatly assisted by his ability to classify rocesses and the details of operations. As far s possible, methods of procedure should be ade standard, this being of importance when e question of cost is considered.

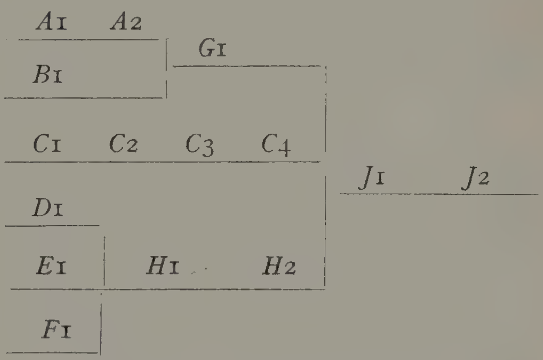
Work to be Done. The planner has to deter-ine the work involved by a careful analysis of ch individual item in itself, and in its relation all the other parts concerned. The results his work are given as a written statement of the sequence of processes to be used for the manufacture of each item from raw material to finished article. The following points have to be considered—

1. *The raw material ordered.* It is necessary to see that the material specified by the drawing office is the most suitable form having regard to the manufacturing processes.

2. *The sequence of processes* that the raw material must pass through.

3. *The details of each process,* such as the type of machine, the tools, equipment, etc. These details will vary very much according to the accuracy and finish required.

Fitting and Assembly. In this class of work the sequence of operations is not so apparent as in the case of machine work. The sequence of assembly operations may be represented by means of a chart, showing the different groups which make up the final article and the order in which they have to be assembled. In the figure given below the letters represent individual groups, and the numbers represent assembly operations on these groups. Thus, the group *A*1 requires a second operation *A*2 before the operation *G*1 can be performed, as *G*1 combines *A*2 and *B*1—



Written Instructions. When the planner has decided on the sequence and details of the different operations he embodies the results in what is usually known as an "Instruction Sheet." This forms a permanent record of the best method known for accomplishing the work, and as a standard, it should not contain any variable details. It should contain such information with regard to the method of procedure that, with the assistance of the corresponding drawing, the operator is supplied with complete information for performing the work in the required way. Instruction sheets may be prepared so as to give complete instructions on one sheet for the manufacture of a part, or each sheet may only contain the details of one operation, the latter probably being the better method.

Details Given. In the case of machine work the following details will be required on the instruction sheet—

1. Name of part.
2. Operation number.
3. Material to be used.
4. Type of machine tool to be used.
5. Detailed description of the operation quoting—
6. Feeds and speeds to be used.
7. Time allowed for each detail.
8. Extra time allowances for resetting the machine, etc.

Advantages of Instruction Sheet. 1. The information recorded is the accumulated experience of many people arranged by specialists, and so is likely to be of a higher level throughout.

2. A standard having been set, the level of work is more likely to be raised.

3. Tools generally will be used more efficiently.

4. The operator's time is saved because there is no need for him to do any planning.

5. The instruction sheet helps to keep the staff efficient because, if everything is not in first-class order, the operator will quickly let those responsible know about it, because the operations are detailed and the times set on the understanding that all the equipment suggested by the planner is available, and in first-class condition.

6. The record is permanent and not dependent on the presence of any particular foreman or workman.

7. The sheets will provide a certain amount of educational benefit for those who care to make use of it.

Specification Sheets. One important document that has not been mentioned so far is the "Specification Sheet"—usually prepared by the drawing office or a separate specification department. When instructions to manufacture are received the specification department prepare these sheets, one sheet or group of sheets generally being used for each main assembly drawing. The essential particulars given are the drawing numbers of each detail part, the name of the part, and the quantity required of every item. What further particulars are given, such as the name of the department manufacturing the part, the name of the department supplying the casting or partly manufactured article, the material used, and so on, will depend on the particular factory's organization of the planning and production departments. By their issue to several departments the sheets may be made to serve a variety of purposes. Thus, if extra money columns are provided, the forms may be used by the cost department for checking up the total material cost of the goods, and by the shipping department to ensure that the whole order has been dispatched. The sheets will also provide the basis of the work of the planning and production departments, giving a convenient summary of all the detailed parts on order. In the case of articles constantly being ordered, the sheets may be duplicated by printing or blue-printing, and then issued as required.

In works where the planning and production departments have not reached a very high state of development, the specification sheets may become very important, and form the authority for the foreman to order up material and generally put the work in hand.

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ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

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LESSON VII

MAGNETISM AND ELECTRO-MAGNETISM

Magnetism and electricity were considered to be separate and distinct sciences right up to the year 1819 when Oersted made a startling discovery, which had the effect of interlinking them together. From then onwards, electricity and magnetism became as one science—*electro-magnetism*—the foundation of the science of electrical engineering.

1. MAGNETISM

A *magnet* is a piece of steel possessing the peculiar property of attracting to it, small pieces of iron. The properties of a magnet may be briefly stated as follows—

1. Will attract small pieces of iron, the power of attraction being concentrated at the ends or poles.
2. If freely suspended in the proximity of another magnet, will take up a definite position relative to that magnet.
3. If freely suspended, will take up a definite position relative to the earth, because the earth itself is a huge magnet.

Many centuries ago, the Greeks were acquainted with the attractive properties of a mineral, notably found in Magnesia, Asia Minor, to which they gave the name "magnet." This natural magnet is known to mineralogists as *magnetite*, and to chemists as *magnetic oxide of iron* (Fe_3O_4). It would appear from ancient records, that as long ago as A.D. 121, the Chinese were aware of the peculiar property possessed by this mineral, of always pointing in one direction (North and South) when freely suspended. The first mention of a magnet needle by European writers was made about the twelfth century. Since then, the magnet (or compass needle) has been utilized for navigation, and for this reason, the mineral magnetite, was named Lodestone (or leading stone) by early pioneers. The natural magnet, Lodestone, has no commercial value as a magnet, since its properties are very weak.

On the other hand, permanent magnets, which are really alloy steels specially heat-treated and magnetized, find innumerable applications in engineering. They are extensively used in electrical instruments of all types, and for small generators of electricity, a very familiar example being the present-day magneto, which can aptly be described as the pulse of the internal combustion engine. As used in the mariner's compass, the permanent magnet has been of untold service to mankind. It is indeed difficult to estimate the far-reaching effect that this small contrivance has had on man's progress towards a better civilization.

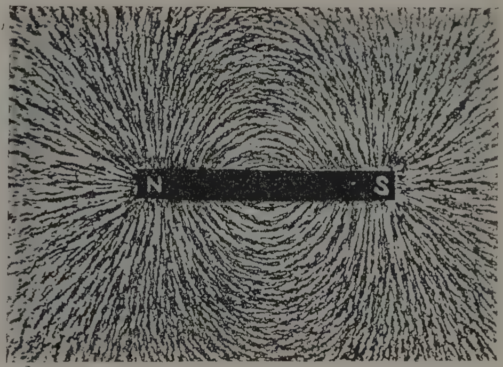


Fig. 30. Magnetic Field of Bar Magnet

(*a*) **Magnetic Lines of Force and Magnetic Field.** Magnets, whether natural or artificial, possess their properties by virtue of the phenomenon surrounding the magnet. If we examine the space surrounding a magnet by means of iron filings, we shall find innumerable forces at work, as evidenced by the order in which the filings arrange themselves. This phenomenon is illustrated in Fig. 30, where we see that the filings, in arranging themselves, trace the paths of forces emanating from the magnet. These forces are referred to as *magnetic lines of force*, and the phenomenon surrounding the magnet as a *magnetic field*. The conception of magnetic lines of force still accepted, we owe to Faraday, who said, "We might suppose forces radiating from a magnet, each individual force being termed a line of force." The grouping of the magnetic lines of force constitutes a magnetic field, and the greater the number of the magnetic lines of force at any given point, the greater is the strength, or intensity, of the magnetic field at that point. The strength, or intensity, of any magnetic field is expressed in lines per square inch or square centimetre, and it is possible very accurately to determine the number of magnetic lines of force in any magnetic field.

(*b*) **Properties of Magnetic Lines of Force.** Despite the fact that these magnetic lines of force are quite invisible, we know in many ways that they permeate the magnet and the space surrounding it, stretching out into infinite space. It can be assumed that each line of force *always makes a complete circuit in itself*, emerging from the N. pole of a magnet, entering again at the S. pole, and completing the circuit through the magnet to the N. pole. Faraday assigned two physical properties to these imaginary magnetic lines of force—

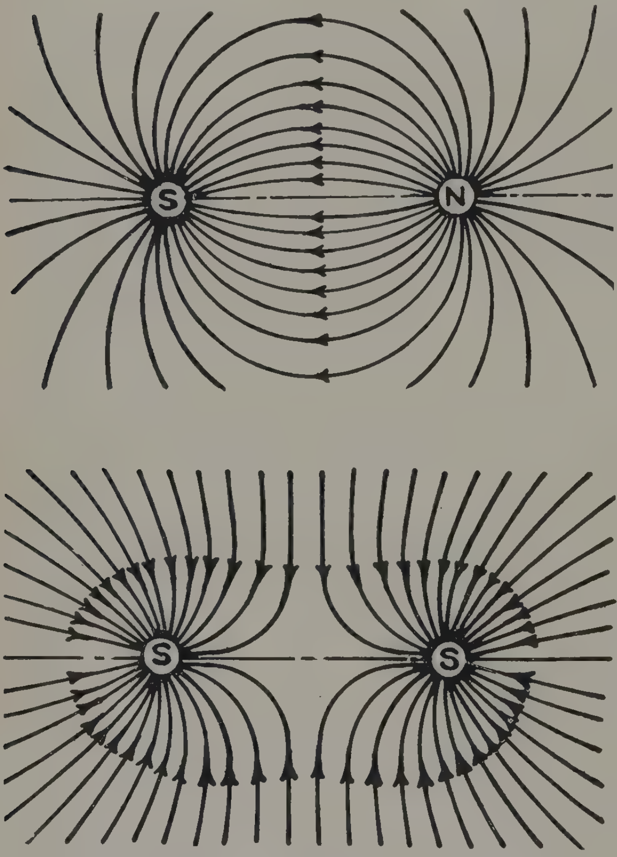


Fig. 31. Magnetic Fields Due to Unlike and Like Poles

1. A magnetic line will always tend to shorten itself from end to end.
2. Magnetic lines when bunched together will tend to repel one another.

Whenever magnetic lines of force are present in space, we may say that a mechanical tension or pull is exerted across the space and in the direction of the length of the magnetic lines. In this respect, magnetic lines of force may be likened to stretched elastic bands, always tending to shorten themselves from end to end. Fig. 31 shows the arrangement of magnetic lines of force for unlike and like poles, and, bearing in mind the properties of these magnetic lines, it will readily be understood why unlike poles attract and like poles repel. In a similar manner, the magnetic field diagram in Fig. 32 will serve to illustrate the attractive force exerted by a magnet on a soft-iron keeper.

(*c*) **The Magnetic Circuit.** The paths followed by magnetic lines of force are called magnetic circuits, for the same reason that the path of an electric current is called an electric circuit. Iron, steel, nickel, and cobalt afford very easy paths to magnetic lines of force, but other materials are not such good magnetic conductors. To denote the relative magnetic conductance of materials, the term *permeability* is used. The permeability of air is unity, whilst that of soft iron may attain as high a value as 3,600. This means that for a given *magnetomotive force*, or magnetic pressure, the number of magnetic lines of force (termed the *magnetic flux*) in a given path, may be as much as 3,600 times greater for iron than for air.

Referring again to Fig. 32, we see that when the keeper is in contact with the magnet poles, the magnetic flux is a maximum. So soon as an air gap is introduced in the magnetic circuit, the number of magnetic lines is considerably reduced, and still further reduced if the air gap is further increased.

The total flux in a magnetic circuit is denoted by the symbol Φ, and if we divide the total flux by the sectional area through which the magnetic lines are flowing, we obtain the *flux density*, which is the number of magnetic lines per unit of area. Thus, the flux density in a magnet can be calculated from the formula—

$$B = \frac{\Phi}{A} \quad . \quad . \quad . \quad . \quad . \quad (17)$$

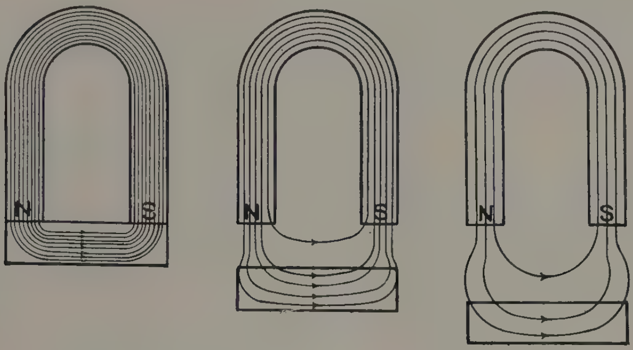


FIG. 32. HORSESHOE MAGNET WITH KEEPER IN THREE POSITIONS

Where, Φ = total flux in the body of the magnet

A = cross-sectional area, in square centimetres.

and, B = flux density in lines per square centimetre.

If the cross-sectional area is measured in square inches, then the flux density will be expressed in lines per square inch.

(*d*) **Non-magnetic Materials.** All metals, with the exception of iron, nickel, and cobalt, are non-magnetic, which means that they cannot be magnetized and so become possessed of the properties of a magnet.

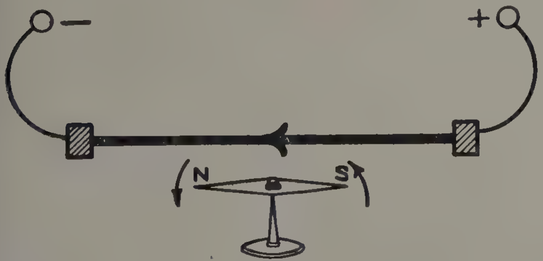


FIG. 33. OERSTED'S EXPERIMENT

It is curious, however, that a 25 per cent nickel-steel alloy is practically non-magnetic, and has a permeability closely approximating to that of air, viz., unity. On account of its great mechanical strength, this alloy is used extensively in certain types of electrical apparatus.

2. ELECTRO-MAGNETISM

(*a*) **Magnetic Field Due to an Electric Current.** Whenever an electric current flows in a conductor, a magnetic field is set up. This very important discovery was made by Prof. Oersted, of Copenhagen, in 1819. He found that if a straight conductor be placed directly above a compass needle, as shown in Fig. 33, and current passed through the conductor, the compass needle is deflected in a direction at right angles to the conductor. This experiment was all the more remarkable, because it was the first time that an electric current had been made to produce mechanical motion.

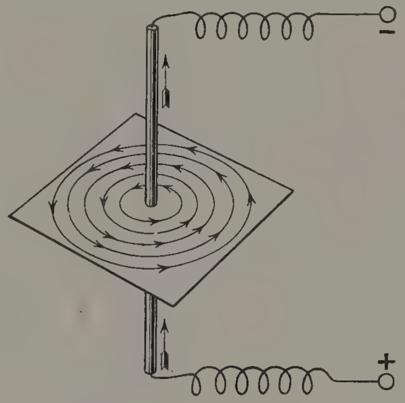


FIG. 34. MAGNETIC FIELD AROUND A STRAIGHT CONDUCTOR CARRYING CURRENT

The reason for the compass needle turning at right angles to the conductor, is due to the fact that a conductor carrying current is surrounded by a magnetic field, in which the magnetic lines are concentric circles around the conductor, as illustrated in Fig. 34. A compass needle always tends to set itself in the direction of the magnetic lines, and in the case of a circular magnetic field, it moves to a position that is tangential to their direction. If the current in the conductor is reversed, the lines of force will flow in the opposite direction, and the movement of the compass needle must, therefore, be in the opposite sense, since the *North pole of the needle will always point in the direction of the magnetic lines of force.*

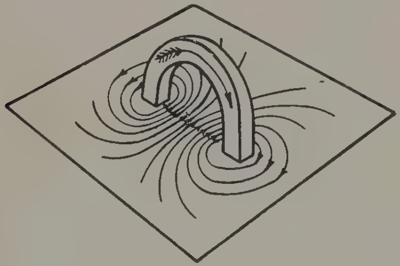


Fig. 35. Magnetic Field of Tangent Coil

(*b*) **Magnetic Field of a Coil Carrying Current.** Now, if we wind the conductor in the shape of a coil, the lines of magnetic force will embrace the coil, in the form of circles passing through the centre of the coil, as shown in Fig. 35. By increasing the length of the coil and making it many times the diameter, we shall produce a *solenoid*, and the distribution of the magnetic lines will be as shown in Fig. 36. It will be seen from the illustration, that the magnetic field is strongest at the centre (*c*), and for a long solenoid, the flux density is practically uniform along its length, except near the ends.

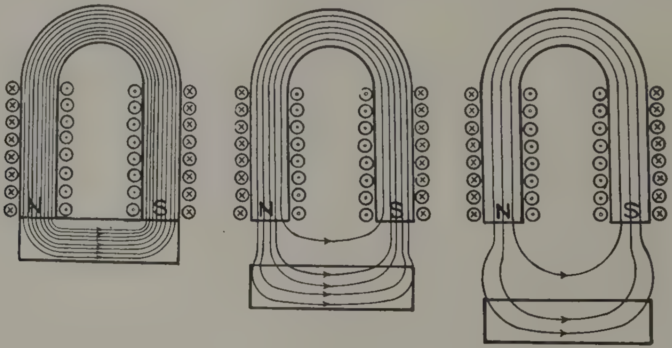


Fig. 37. Horseshoe Electro-magnet with Keeper in Three Positions

(*c*) **The Electro-magnet.** A soft iron core inserted in a solenoid, as shown in Fig. 36, would greatly increase the magnetic flux, since iron affords a very much easier path than does air. In other words, the permeability of iron is many times greater than that of air. So long as current is flowing in the coil, the soft iron core behaves in precisely the same manner as a magnet. The polarity of the iron core would also be reversed on reversing the current through the coil.

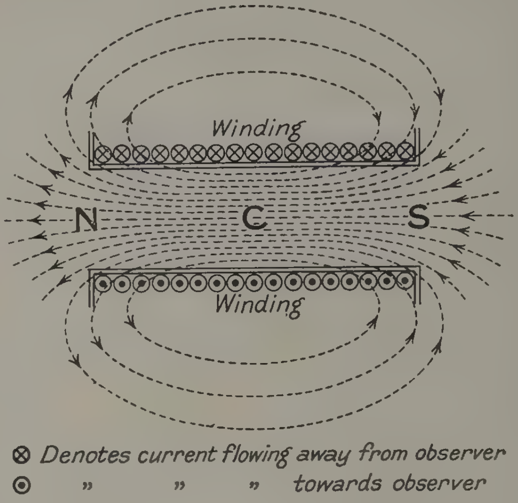


Fig. 36. Magnetic Field of a Solenoid

It is helpful in remembering the relationship between the current and magnetic field directions, to visualize the current as moving in the direction of the point of a corkscrew when it is turned, the direction of turning the corkscrew corresponding to the direction of the magnetic field. Thus, for a clockwise movement of the corkscrew, corresponding to the magnetic field direction, the current would flow in a forward direction (represented ⊗), and for an anti-clockwise movement, the current would flow backwards (represented ⊙). This is known as the *Corkscrew Rule.* By imagining the point of the corkscrew on any one of the conductors in Fig. 36, and assuming it is turned in the direction of the magnetic field, the student will be able to verify this rule.

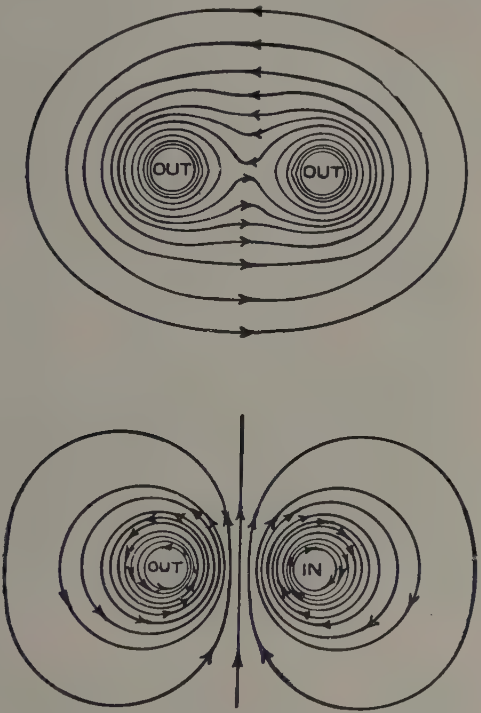


FIG. 38. MAGNETIC FIELDS DUE TO PARALLEL CONDUCTORS CARRYING CURRENTS IN TWO DIRECTIONS

Fig. 37 shows three flux diagrams for a horse-shoe electro-magnet and keeper, and it is of interest to compare these diagrams with those shown in Fig. 32. The flux distribution is precisely the same, and whilst current is flowing in the coils, the soft iron core has all the properties of a magnet.

The electro-magnet has a wide field of application. Very powerful electro-magnets, capable of lifting many tons of iron, are used very largely in engineering. Electro-magnetic chucks are extensively employed in machine shops, and in almost every home we find the electro-magnet in use, in the form of an electric bell.

3. ELECTRO-DYNAMICS

Electro-dynamics deals with the forces generated by the interaction of a current carrying conductor and a magnetic field.

(*a*) **Forces Between Parallel Conductors Carrying Current.** If two parallel conductors are carrying current in the same direction, the magnetic lines of force extend themselves to embrace the two conductors, and in so doing, a force of attraction is exerted between the conductors. The upper diagram of Fig. 38 shows two such conductors, and by the aid of the elastic band analogy it is easy to understand how this attractive force is generated. Now, if the conductors are carrying current in opposite directions, there will be a force of repulsion, as indicated in the lower diagram in Fig. 38. It will be observed that in this case the magnetic lines of force between the two conductors are, owing to their mutual repulsion, constrained into a smaller space, and in consequence, the conductors repel one another.

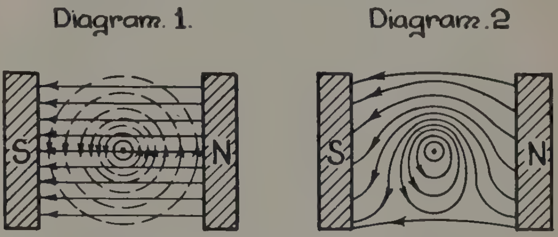


FIG. 39. STRAIGHT CONDUCTOR LYING IN MAGNETIC FIELD

(*b*) **Force Exerted on a Conductor Carrying Current in a Magnetic Field.** A conductor carrying current, when placed in a magnetic field so that the axis of the conductor is at right angles to the direction of flow of the magnetic lines, is acted upon by a force urging it in a direction at right angles to the magnetic lines. This will be clearly understood by reference to Fig. 39.

In Diagram 1 we see the two separate magnetic fields due to the magnet poles and the current, whilst Diagram 2 shows the resultant magnetic field. It will be clear that the conductor, owing to the distorted shape of the magnetic field, is acted upon by a force, urging it in a downward direction. On reversing the current, the force exerted will operate in an upward direction. This is the principle of the electric motor, with which we shall deal more fully later.

PIONEERS OF ENGINEERING

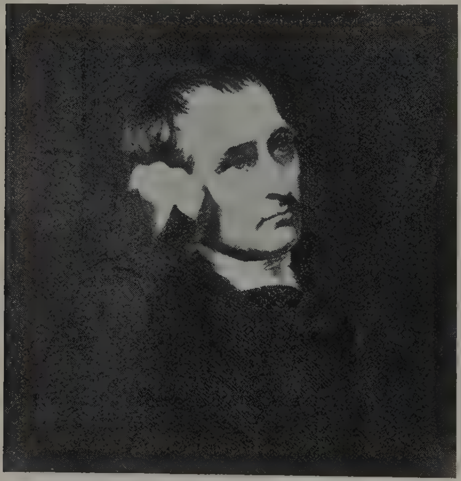
By J. F. Corrigan, M.Sc., A.I.C.

LESSON XIV

THE "MULE"

SAMUEL CROMPTON (1753-1827)

Arkwright's invention of the spinning jenny had, as we have already seen, a tremendous influence upon the cotton industry of England. But Arkwright was by no means the sole textile inventor of the period. Another name, almost equally brilliant as that of Arkwright in the annals of textile invention, is Samuel Crompton, of Bolton, the far-famed inventor of the "Mule."



Samuel Crompton

Born near Bolton in 1753, Crompton was from his earliest years imbued with the prevailing technique of the cotton industry. Bolton was the seat of cotton manufactures, and Crompton, born into a family which for generations had carried on the industry of the hand loom, was at a very early age apprenticed, if we may use that term, to the trade.

Crompton, however, was not like other hand weavers. He possessed a mechanical genius, and, encouraged by the success of Kay, Arkwright, Hargreaves, and others, he set to work in order to produce a machine which would spin yarn without the tedious hand operations which were necessary at that period.

And so the inventor eventually produced his "Mule." There is an aspect of romance about the invention of the mule. It was first got into working order at a lonely spot near Bolton, Lancashire, known as Hall-in-th'-Wood. The machine was completed in the year 1779. It was unpopular with the general run of hand operatives, and this on account of the fact that the local workers saw in it a means whereby their labour would be discarded, and their source of income taken away from them.

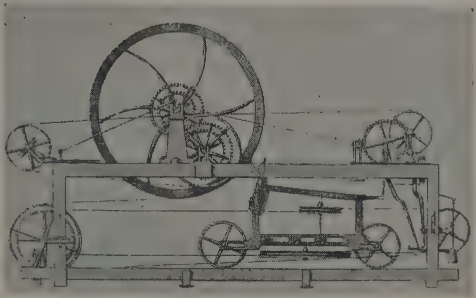
Hall-in-th'-Wood, near Bolton, where Crompton Invented the "Mule"
(From an Old Engraving)

For a time, therefore, Crompton kept his invention a fairly close secret, showing it only to a few interested persons at the lonely house at Hall-in-th'-Wood. Various stories are extant about the visitations which Crompton received from the local workers, who did everything possible to destroy his machine.

But the mule was a far too valuable invention to be kept a secret. Enlisting the services of interested people, Crompton placed his invention at the disposal of the cotton industry in general. He did not patent the machine, preferring to trust to the generosity of his fellow men after they had once fully realized the benefits which the mule would impart to the industry as a whole.

Crompton's mule was universally adopted in the cotton industry, and, at a later date, Parliament voted the inventor the sum of £5,000 as a reward for his machine. Such a vote, however, was not carried until it had been pointed out that Crompton's invention had resulted in the yearly addition of £350,000 to the country's revenue.

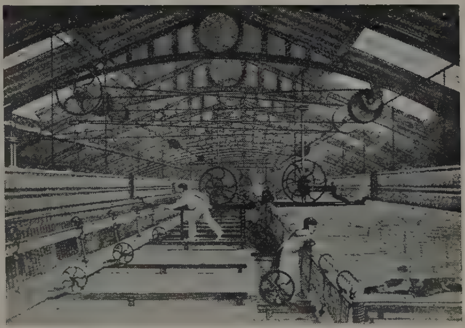
Crompton, however, soon spent this money in the furtherance of other inventions, and when he died in 1827 he was a comparatively poor man.



An Early Spinning Mule

In its essentials, the mule consists of a system of rollers and an arrangement of spindles by means of which the rough cotton yarn is first drawn out and then given a twist, which process serves to consolidate the structure and strength of the thread. The cotton thread is then automatically wound on bobbins, whereupon the drawing out of more of the rough cotton is repeated. This again is twisted and wound and thus the process is repeated automatically, and with a nicety which was impossible of attainment by means of the earlier hand methods.

Crompton's " mule " led to the production of very fine spun yarns, and it was not long before the device was applied to the production of linen and woollen yarns also. The mule, of course, speeded up the production of yarns enormously, and when steam power was applied eventually in order to drive the machines, so great was the production of cotton yarns that Lancashire became the home of the world's cotton goods, a position which that county has held almost to the present day.



The " Mule " in Operation

GAUGING AND INSPECTION

By Joseph G. Horner, A.M.I.Mech.E.

LESSON I

RULES, CALIPERS, AND SURFACE GAUGES

Behind this subject lies the long story of the evolution of standards of measurement, and the tedious preparation of national bars for reference. The instruments used by the mechanic are based on these, either the yard or the metre, translating the long labours of the scientific pioneers into the enlisted service of the workshops.

The Classes of Measurements. These include linear dimensions—lengths, widths, thicknesses, diameters, and that of angles. By far the largest volume of work done, and the number and variety of instruments used, are concerned with the first. These include direct observation, as, determining a dimension by placing it in contact with the lines on a rule, or that of the gauge, in which the piece to be measured is brought into direct contact with surfaces on the instrument, which embrace it, or by which it is embraced. Of these there are two great groups—those in which the dimension is rigidly fixed and permanent, as in the fixed gauges, and those where it is capable of variation, as in the common caliper, the beam calipers, the micrometer, and vernier calipers, and the standard measuring machines. Angles are measured with common bevels, with bevel protractors, and with fixed gauges.

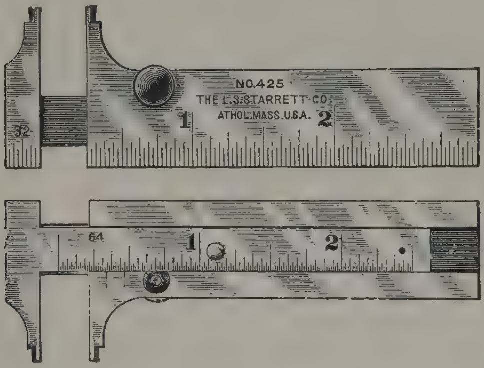


Fig. 1. Slide Caliper

Measurement by Sight and by Touch. It may appear rather contradictory, that while the fixed gauges have long superseded the use of the common rule and calipers for precise work, yet the finest measurements are still made by direct readings of divisions. The explanation is, that the aid of the micrometer, or of the vernier, supplements the main element, the screw of fine pitch, or the rule. $\frac{1}{100}$ in. is difficult to measure on a rule, but from $\frac{1}{10000}$ to $\frac{1}{50000}$ in. can be detected on the other forms, and still finer dimensions on some of the measuring machines.

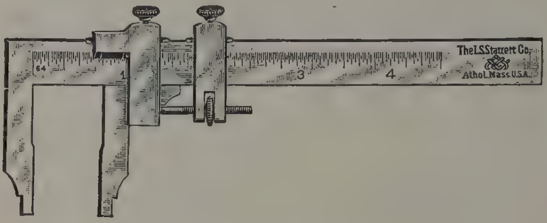


Fig. 2. Caliper Square

The Common Rules. These are still indispensable for all ordinary and general work, and they are now admirably made with graduations to suit all requirements. A usual length is 12 in., others 6 or 3 in. long go in the waistcoat pocket. Many are fully divided along their whole length, down both edges, and many on both sides. Graduations in one may include $\frac{1}{32}$ in. and $\frac{1}{64}$ in., another $\frac{1}{64}$ in. and $\frac{1}{100}$ in. English and metric dimensions may be included on the same rule, or be confined to separate ones. Special rules are made for gear cutting proportions, graduated decimally, or in vulgar fractions. Some rules are graduated across one end to measure depths and widths of grooves and recesses. A hook is fitted quickly on some rules to take a measurement from a shoulder, to be inserted in a small hole where the end cannot be set by sight. Rules are tempered rigid, and flexible. A few are now made of stainless steel.

Calipers and Compasses. In all the ordinary run of work that does not include interchangeability, the rules are in constant request for direct measurements, being laid against the work for that purpose. But in an equally large number of cases the aid of an intermediary instrument, either the caliper or compass, is enlisted. Each occurs in dozens of variations of form. They are set as finely as practicable by the divisions on the rules, to be transferred to work in progress, or measurement taken from the work is checked on the rule. But in each case nothing finer can be attempted than that which is afforded by visual observation of the coincidence of caliper or compass with the divisions on the rule.

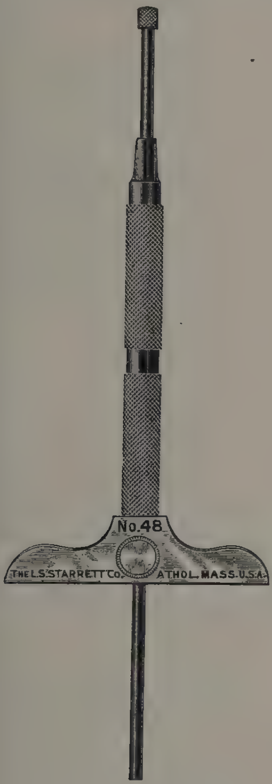


Fig. 3. Depth Gauge

Contact and Pressure. The sense of touch or contact arises when calipers are used. Here uncertainty enters, due to the difference in the degree of this sense in the hands of delicately or clumsily fingered men, which will show marked differences in measurements. Another cause of error lies in the calipers, which should not be flimsy or inclined to spring. If the legs spring they yield to slight pressure and will not register exactly. Bare contact alone is essential, and this is not easily controlled. In the micrometer tools and measuring machines, trouble is apt to arise due to differences in degrees of pressure, and measures highly elaborated in some of the latter are taken to control its degree.

All calipers are well tapered from the joint. The pins must fit closely without slop, the points are narrow to give very restricted surface contact. They must be held, not by the legs, but at the joint. The larger sizes should be held vertically, for, if horizontally, the weight of the legs will vitiate results. The heavier the calipers are, the greater is the liability of the legs to spring. Minute adjustments of the legs are effected by giving gentle taps to one of them. The joint holds frictionally, no clamping being required as in compasses, but large numbers are made with a bow, like spring dividers, adjusted with a screw and milled circular nut. Calipers must be held at right angles with the work, for a slightly diagonal setting will prove incorrect. On large diameters, a little lateral movement will be necessary to find the square position, which is that of the easiest fitting. Calipers measure external and internal diameters, and either may be transferred to, and compared with, the divisions on a rule. To check one from the other, external and internal are brought into contact, as when a shaft and a bore have to be mutually tested. Some forms are made to avoid this, by combining two pairs of legs, external and internal with a central pivot common to both. Other variations in forms for special services are very numerous.

Caliper Rules. These are representative of a group of instruments that combine caliper and other measurements with rule divisions—sliding calipers (external and internal), depth gauges. Caliper rules, beam calipers, slide calipers, caliper squares, as these instruments are variously termed, have the advantage of a double contact, between a fixed and a movable jaw, set by the graduations that are common to rules. A plain slide caliper is shown in Fig. I, having a movable slide marked in $\frac{1}{64}$ in. on the slide and $\frac{1}{32}$ in. on the stock. It is clamped with a knob and left-hand thread, using

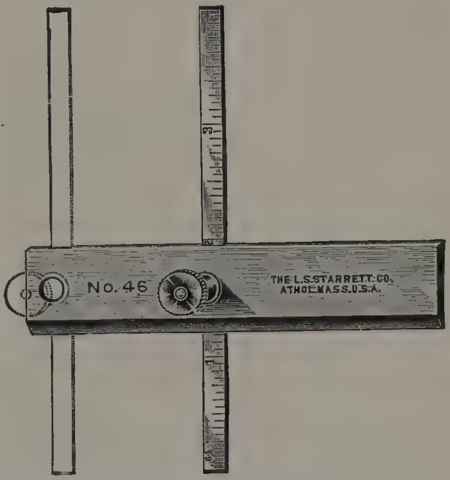


Fig. 4. Depth Gauge in Which the Scale Can be Clamped Close to One End

the thumb of the same hand as that in which the instrument is held. Two lines on the stock, seen in the lower view, enable inside and outside measurements to be read. In another of the same type, graduations are given to read the circumference that corresponds with diameter, being useful to see at a glance the suitable surface speed for turning or boring. The finer instruments are made with two movable heads, one being a setting or abutment head, slid and clamped in a convenient position, to adjust the measuring jaw minutely with a milled-headed screw. A caliper square of this class is shown in Fig. 2. Inside dimensions are taken with the outsides of the nibs on the jaws.

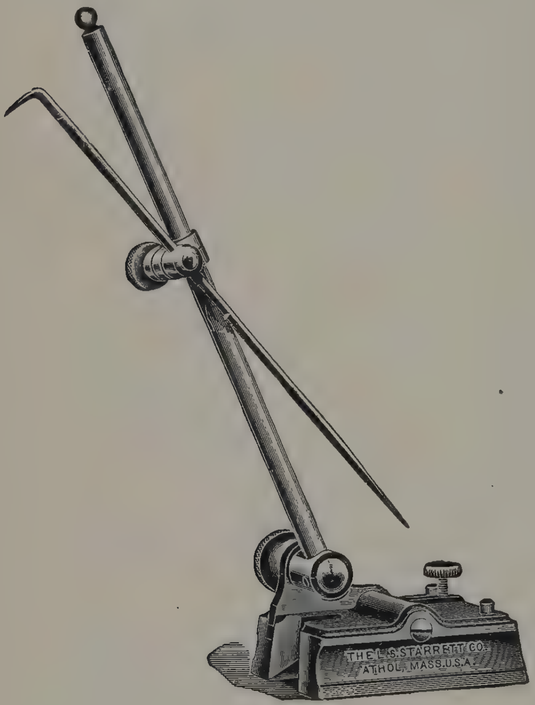

THE L.S. STARRETT CO.
ATHOL, MASS. U.S.A.

FIG. 5. SURFACE GAUGE

Depth Gauges. Depths are measured with the brass tongue of a slide rule, or by the insertion of a common rule, or by means of internal calipers set between a straight-edge and the bottom of the recess. Better methods are available by the employment of narrow rules sliding in heads. The head bridges the recess, and the depth is measured with the wire or the rule slid down until it touches the bottom, when it is clamped. Fig. 3 shows one of this kind. A spring in the barrel automatically forces the rod downwards. It is held in the groove of the head with a knurled lock nut. In another design, Fig. 4, the head is lengthened, to permit of inserting the rule close to one end, in situations that would not admit the head in Fig. 3.

Surface Gauges. A necessary instrument that is preferable to the rule is the surface gauge or scribing block. Its extensive functions cannot be fulfilled by any other tool. It will mark lines parallel with a horizontal base or a vertical edge, and on surfaces that are not on the same plane, and around irregular outlines, and all within an extensive range of dimensions. It will describe lines on upper and under surfaces, and gauge heights, and transfer heights and distances from one face to another. It is made to slide on plane surfaces or along edges, or to straddle circular parts.

The general outline of a surface gauge is that of a heavy base supporting a cylindrical pillar on which a split block, adjustable vertically, carries the scribing rod movable through it. The block that receives the rod and in which it is clamped, swivels to set the rod at any angle. The instrument, therefore, provides for positioning for various heights, horizontal distances, and angular settings. One end of the rod is a straight point to deal with vertical faces, the other is turned round to a right angle to mark or gauge on horizontal surfaces. Around this general design there are many modifications.

Fig. 5 is an example where the base is grooved through the bottom and end for use on circular work as well as on flat faces. Gauge pins at the rear are used to bear against the edge of a surface plate, or the slots in a planer table. The pillar can be set at any angle in a rotating head jointed to a rocking bracket pivoted in the base, adjusted with a knurled screw at one end acting against a stiff spring at the other. These instruments are made by the L. S. Starrett Co., Ltd., of Upper Thames Street, London, E.C.4.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON II

CAST IRON

The Remelting of Iron in the Foundry

The remelting of pig-iron for castings is conducted in any of the following types of furnaces—

Crucible,
Reverberatory,
Cupola,
Open hearth.

Of these, Nos. 1 and 4 will be dealt with in subsequent lessons on the manufacture of steel.

The Reverberatory Furnace. In this type of furnace the materials to be melted are not in actual contact with the solid fuel, but only with the flame and products of combustion. Built of firebrick, encased in iron plates, the reverberatory furnace consists essentially of a hearth to contain the charge, and a fire chamber wherein the fuel is burnt. These two chambers are intercommunicating but partially separated by a firebrick bridge which extends almost to the slightly arched roof. The roof slopes downwards from the fire bridge to the flue bridge and the flames, striking this, thus "reverberate" on to the charge. The products of combustion are drawn away by a chimney stack fitted with a damper for purposes of temperature and draught regulation.

The use of this furnace in foundry practice is more or less confined to the melting of iron for such purposes as chilled rolls, etc., and to a less degree, malleable castings—where absolute uniformity of quality is essential. In this connection it should be noted that a characteristic peculiar to foundry reverberatories is the double-arched roof, which dips suddenly over the centre of the hearth and thus accentuates the downward sweep of the flames.

South Staffordshire coal is a favourite fuel, as it yields a long smoky flame and is comparatively free from sulphur. The stoking door is kept as small as possible, and a thick bed of fuel is maintained, so as to prevent the creation of oxidizing conditions in the furnace.

Remelting by these means is, however, a costly proceeding and, in this respect, at least, cannot compete with the cupola.

The Cupola. This is, by far, the commonest type of melting furnace for foundry practice. It may be likened to a small blast furnace, consisting as it does of a vertical cylindrical shaft, firebrick lined and supplied with a tuyere, or tuyeres, through which the air necessary for the combustion of the fuel is blown. The coke, along with the broken pig, scrap metal, and flux, is charged from near the top and the molten metal and slag drawn off, as required, from the bottom.

The object of the flux—chiefly calcareous—is to unite with the silica in the sand adhering to the pig-iron, and the coke ash, thus making a fluid slag that can be tapped off when necessary.

Cupolas may be broadly classified as belonging to one of two types—*solid bottom* and *drop bottom.*

In the former the breast of the furnace, being made removable, is taken away on completion of the melting and tapping operations, the lining is broken down and the red-hot coke and other debris remaining raked out. Careful preparation of the bottom is then obviously necessary before another run can be started.

The distinguishing characteristic of the drop bottom cupolas obviates the necessity for much of this lost time. The furnace is supported on columns, ample working room being left between ground level and the bottom, which is made up of two hinged portions. Previous to a melt these are keyed together and faced up with moist sand to a depth of about 3 in., and further supported by props. After the last cast has been made the two portions are let down and the unburnt coke, the residue of the iron, slag, etc., "dumped." Any metal remaining after tapping can then be cooled off, broken up, and used in a subsequent charge.

It should be remembered, when comparing the working of the cupola with the blast furnace, that the latter involves the reduction of the iron from the ore, whereas, in the former, the iron is already in the metallic state and has, therefore, no need of a strongly reducing atmosphere. Now, when carbon is burned to carbon monoxide the heat units given out only total about one-third of those given out when carbon is burned to carbon dioxide; therefore, if blast furnace conditions are reproduced and an excess of CO formed, this, whilst creating a reducing atmosphere, also results in a considerable loss in heat value. Conversely, if perfect combustion occurs and CO_2 is formed, an oxidizing atmosphere is created. This, if permitted, would have the obvious effect of oxidizing the metal, which must, at all costs, be avoided. Hence, considering fuel economy on the one hand and the quality of the metal on the other, it would appear that the most efficient atmosphere for cupola practice is one of a *slightly* reducing nature.

From this it follows that tuyere area, blast pressure, and the coke : iron ratio are all of the utmost importance.

Blast pressures vary according to the dimensions of the cupola; the blast must reach the centre of the descending charge to ensure regular working and even melting. Thus, only working conditions and experience can fix the ratio of tuyere area to furnace diameter, but, as regards pressures, 5 and 16 oz. per sq. in. may be cited as respective extremes in British practice.

The distance between the cupola bottom and the tuyere level regulates the quantity of molten metal a furnace will hold and so, in small solid bottom cupolas from which heats of various sizes are required, it is necessary for this distance to be made adjustable. This is effected by having two tuyeres, one placed somewhat higher than the other. Only one of these tuyeres is used at a time, the other being blocked up.

With bigger furnaces two, or more, tuyeres are arranged round the furnace. The next step is the introduction of double and triple rows of tuyeres, by which means a more complete combustion of the carbon monoxide is obtained; and examples of elaborations on this arrangement in the way of a spiral series of tuyeres can be instanced.

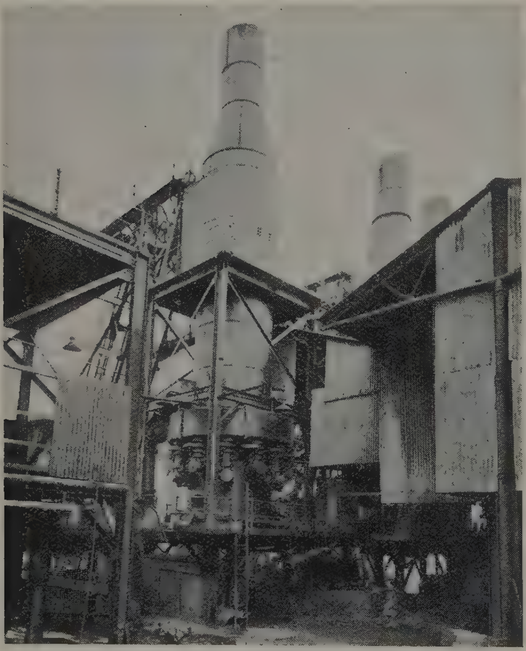


FIG. 1

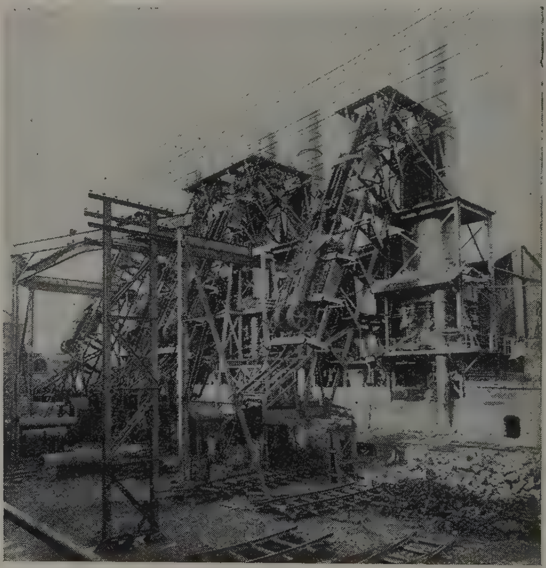


FIG. 1A

MODERN CUPOLA PLANT

These photographs illustrate plant consisting of four 15 tons per hour cupolas. They have been specially prepared and are reproduced by kind permission of The Stanton Ironworks Co., Ltd.

As an example of a typical British cupola, the "Stewart Rapid" may be mentioned. This furnace, which is designed with a double row of tuyeres, has a reduced diameter in the melting zone, and an additional feature in the form of a "receiver" for the molten metal. This is a refractory lined chamber, circular in plan, situated in front of, and slightly below, the tap hole. Its function, as is implied by the name, is to receive the molten iron ready for casting. To assist in keeping this well up to temperature a gannister lined pipe is fitted between the cupola and the receiver, and thus affords a means of hot air supply. This cupola is driven with forced draught in the ordinary way, the blast being somewhat warmer than the atmosphere, the tuyeres being connected to an air belt which encircles the furnace.

Bearing in mind that, as a general rule, cupolas are worked on a series of daily runs, as distinct from the infinitely larger scale of operations in the blast furnace which is kept in blast for a period of years, the charging arrangements of the two have much in common as to rotation of materials, etc. All materials entering the cupola are, or should be, weighed and the weights of the individual charges of pig (previously broken to convenient size), scrap metal, and coke carefully prearranged. In this connection no hard and fast rule can be laid down, as everything depends on working conditions and the individual cupola. These, and the knowledge which only comes of long experience, alone can formulate the important ratio of fuel charged to iron melted. Professor Turner, however, quotes a general figure of 1¼ to 2½ cwt. of coke per ton of iron melted, this being greater with small outputs.

The Influence of Remelting on the Iron. It is the iron which comes *out* of the cupola which is the chief concern of the ironfounder and so, before he can calculate what to put in, he must know what happens to the metal during the melting. Fortunately, remelting has but the slightest effect on the chemical composition of the iron charged in the cupola. The most important change is in regard to the percentage of silicon, which, in a grey iron, will decrease by about 0·2 to 0·3 per cent. The loss in manganese is of a like amount if the original content is in the region of 1·0 per cent, but becomes negligible with smaller initial percentages. Sulphur shows a gain as a result of absorption from the coke. This gain is the more pronounced as the percentage of that element in the fuel is higher and is also increased by cold working. A 0·02 to 0·04 percentage rise can be given as the minimum to be expected with the usual qualities of foundry coke. The total carbon content does not change to any appreciable extent.

The Calculation of Cupola Mixtures. Proceeding with the calculation of the weights of the various qualities of pig-irons to be used as a mixture to make a cast iron for a specified purpose, it is as well to work backwards.

The purpose of the casting suggests to the practical ironfounder the ultimate chemical composition. Its dimensions are also definite, and these, in an ordinary sand or loam mould, determine the rate of cooling. From this, then, the final percentage of silicon can be fixed and the correct ratio between the free and combined carbon thus controlled. The metal must be kept as free as possible from sulphur. Correction must then be made to these two figures to allow for the changes in melting already indicated, due notice being taken of the sulphur in the coke to be used. The data thus obtained will furnish sufficient guide for the selection of iron from the stocks on hand.

NOTABLE EVENTS IN ENGINEERING HISTORY

By E. A. Forward, A.R.C.S., M.I.Mech.E.

LESSON VII

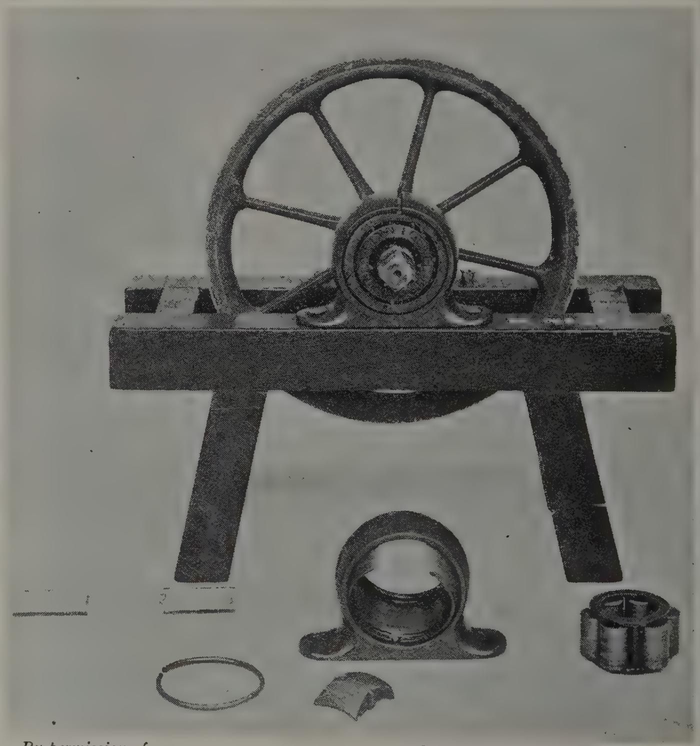
POWER TRANSMISSION

Before the introduction of the rotative steam-engine, the transmission of power to any distance was little required as the prime movers were directly connected with the machinery to be driven, so that primitive wooden shafts and gearing sufficed. (See Lesson III, Fig. 10.) With the advent of more powerful motors, however, several machines could be driven, and means for connecting them with the source of power increased in importance. Shafts were square or polygonal balks, turned only where they rested in their bearings, while the turned parts were sometimes plated with iron strips. In other cases iron pins were let into the ends of the shaft to form the journals. About the middle of the eighteenth century, Smeaton introduced cast-iron shafts, usually of square section to facilitate the attachment of wheels. Later, wrought iron was used, which permitted higher speeds with economy in weight, and subsequently mild steel.

The bearing, at first a simple box of wood or metal, was improved by the provision of renewable brass bushes to take up wear and reduce friction. The lining of the bushes with anti-friction metal, first brought out by Babbitt in 1839, and means for horizontal and vertical adjustment followed, while finally the mounting of the bearing on a ball joint enabled it to adjust itself to the shaft. Where flat surfaces are in contact, the difficulty of maintaining an oil film between them was solved by the rocking-pad bearing invented by A. G. M. Michell in 1905, which enables such bearings to carry ten times the normal load.

The method of reducing friction by mounting a shaft on the peripheries of narrow wheels of large diameter, was introduced by H. Sully about 1720. The more generally applicable method of interposing a number of small rollers between the shaft and the bearing was patented by Garnett in 1787 (Fig. 27), but it is only within recent years that such bearings have been rendered satisfactory for heavy loads. Ball bearings, though proposed in the eighteenth century, received their first important application, from 1876 onwards, in cycle construction, and their use has been gradually extended. These advances in both roller and ball bearings have been rendered possible by the adoption of more suitable materials, and by the development of machinery for manufacturing the component parts with the necessary accuracy.

Lengths of shafting in line were at first coupled together by a keyed box or muff, and



By permission of the Director of the Science Museum

Fig. 27. Garnett's Roller Bearing, 1787

later by bolted flanges, while slightly flexible couplings were introduced to allow for errors in alignment. For disconnecting one shaft from another a sliding jaw coupling can be used at low speeds, while for high speeds a frictional engagement is more suitable, which is obtained in cone, expanding ring, or plate clutches. For coupling intersecting shafts not in line some form of universal or Hooke's joint is used. Where a machine has to be moved about while working, a flexible shaft, which may consist of a wire rope, an articulated shaft, or a helical coil rotating within a flexible tube, is employed.

For connecting shafts not in line and running at the same or at different speeds, toothed gearing was very early adopted, the large wheels being of wood with inserted teeth while pinions were formed of round pins fixed in discs. (See Lesson III, Fig. 11.) Toothed wheels of cast iron were introduced by John Rennie at the end of the eighteenth century, but inserted wooden teeth were still largely used, as they were more easily formed and reduced the noise. Bevel wheels with tapered teeth appeared at the same time. The properties of cycloidal teeth were worked out by Camus in 1752, and of the involute tooth by Euler in 1760. More efficient gearing was then produced, and the cycloidal form was for long most favoured. Machines for cutting teeth had been used in the seventeenth century for clock wheels, but for heavier work they did not appear until the beginning of the nineteenth century. Subsequently, the use of formed cutters for milling teeth from solid blanks, and the ease with which the involute tooth can be generated has led to the universal adoption of that form. Wheels with stepped teeth were in use as early as 1775, and these have developed into the helical form by which great smoothness of running is secured.

The rope or belt drive which transmits power by frictional contact is not positive, but is flexible and silent. The round belt with a grooved pulley is of early origin, but the flat belt appears to have come into use about 1800, and jockey pulleys were employed to give the

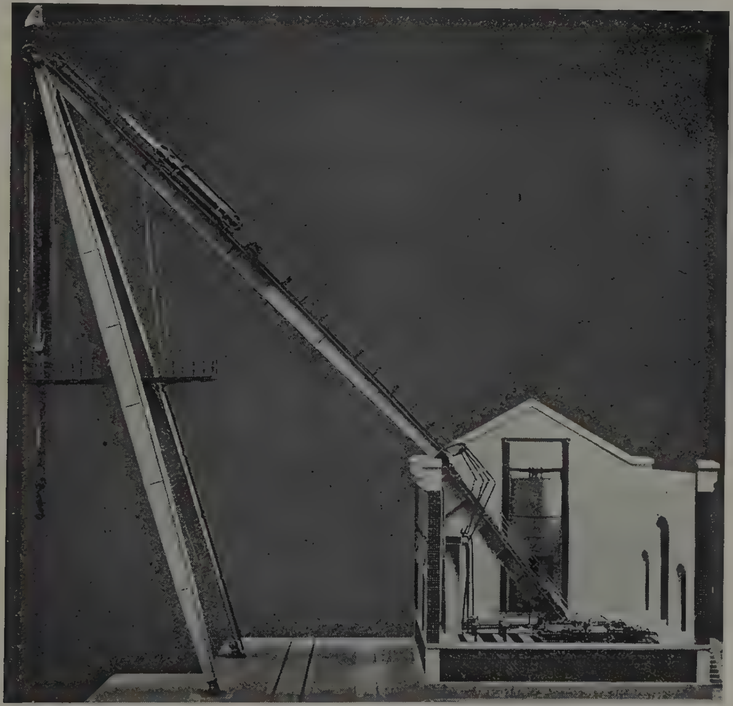


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FIG. 28. HYDRAULIC SHEER LEGS

tension and to form a simple means of disconnection. Leather was at first used, but its high cost led to the introduction of different kinds of fibre belt. For longer drives and higher speeds the rope drive is more suitable and came into use for factories about 1850, cotton being the material generally employed. Wire ropes have been used for mine haulage since about 1830.

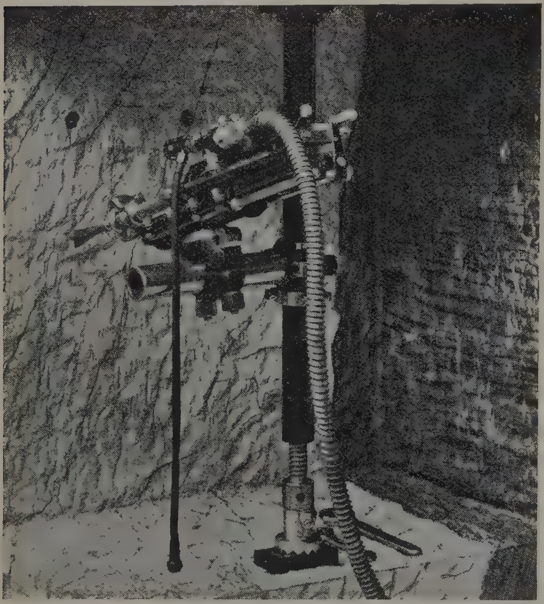
FIG. 29. COMPRESSED AIR ROCK DRILL

Pitch chains have the advantages of belting with the positive transmission of toothed wheels. Simple block chains were used in the eighteenth century, but the modern development began with the roller chain invented by J. Slater in 1864. The improved bush form was brought out in 1880, and the "silent" chain with projecting teeth in 1885 by Hans Renold.

Hydraulic power transmission was schemed by Bramah in 1802, but was not generally applicable until Lord Armstrong had constructed his first hydraulic crane in 1846, and invented the hydraulic accumulator in 1851 (Fig. 28).

Power was transmitted by compressed air early in the nineteenth century by Wm. Murdock. Since its adoption by Sommellier, in 1861, at the Mont Cenis tunnel, the method has been extensively used in mining and tunnelling (Fig. 29), while latterly it has been largely applied to the driving of portable tools.

The most notable advance in power transmission has been made by the use of the electric current by which power can be conveniently transmitted over long distances with little loss. A dynamo, conductors, and a motor, comprise the means for connecting the prime mover with the machine to be operated, and, while the system involves a double conversion of energy, the electrical losses are so small that it can compare favourably with mechanical transmission; at the same time its advantages in general convenience are enormous (Fig. 30).

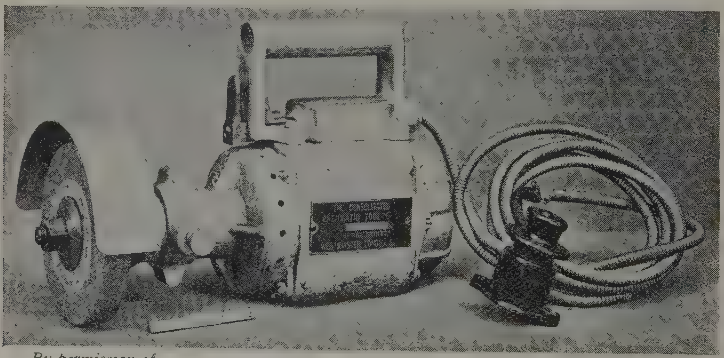
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FIG. 30. PORTABLE ELECTRIC GRINDER

SPECIAL STEELS

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PART II

TYPES OF ALLOY STEELS AND THEIR PRACTICAL APPLICATIONS

(*Continued from page* 636)

(*c*) **High Tenacity Steels.** These are mainly of the chromium, nickel, or nickel-chromium type, these elements being added in comparatively small proportions, i.e. not more than a few per cent, and varying according to the purpose for which the steel is intended. The addition of chromium gives increased hardness, and, in conjunction with nickel or vanadium, gives alloys which are exceptionally strong and resistant to wear and yet can be machined easily. In all cases, correct heat treatment is essential to the development of the advantageous properties of these steels.

Various high tenacity steels are essential in the construction of aeroplanes and motor vehicles as, for instance, in the axles, crankshafts, bearings, gears, and other parts where high tensile strength combined with resistance to shock and fatigue is needed.

Fig. 3 shows the remarkable toughness now obtainable with steel of a quite high tenacity. The bar shown, which is of heat treated steel known as "Hecla 71," has a Brinell hardness number of 410; it is 1⅛ in. in diameter and was bent double cold without fracture over a presser of 1¼ in. diameter. A tensile test piece taken from an adjacent portion of the same bar gave the following results: yield point, 86 tons, maximum stress, 91 tons per sq. in., elongation, 13 per cent, reduction of area, 47 per cent.

Full use has also been made of the advantages of these various steels in war material. Indeed, the development of alloy steels for purposes of peace has been largely assisted by the knowledge obtained from their application to such articles as armour plates and armour piercing projectiles, guns, and gun shields, bullet-proof plating and torpedoes.

Many of the largest single articles yet made in alloy steels have been produced in connection with war material; but composite products, such as tramway track layouts in manganese steel, built up from a number of separate parts, often exceed 100 tons.

The large steel casting shown in Fig. 5 is representative of a number of armour structures used in warship construction, which include besides ammunition tubes such as that illustrated, gun shields, turret roofs, and director towers. These are of special composition and heat treatment, and offer remarkable resistance to shell fire even under concentrated attack. The possibility of producing structures of such excellent quality, in the form of castings in one piece, brings an added advantage, in eliminating weakness due to joints which would necessarily be present in a structure built up from forged parts. The casting illustrated has been attacked by projectiles of several types,

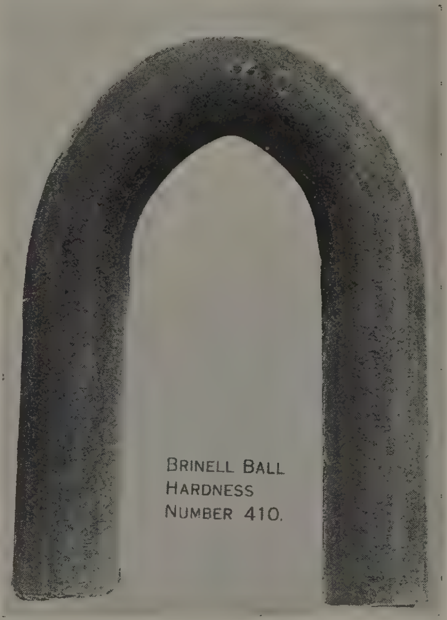

Fig. 3. Bent Bar

Illustrating the toughness of modern high tenacity stee
(90 tons tensile)

including armour piercing up to 9·2 in. calibre; the total striking energy expended upon it amounted to 50,000 ft.-tons, with individual impacts up to 11,000 ft.-tons. The tube has successfully withstood this severe test without a sign of a crack.

(*d*) **Non-corroding Steels.** The author has estimated approximately that the total world wastage, due to corrosion of iron and steel, amounts to about £500,000,000 per annum. The losses largely occur through the familiar forms of rusting under the action of air and water. There is thus here a great potential field of utility for special steels.

The introduction of 12 to 14 per cent chromium steel, sometimes called non-corroding and sometimes stainless steel, also with comparatively low carbon, up to about 0·30 per cent, did much to encourage the hope of substantial success in fighting these ravages of corrosion by means of alloying. Time-honoured methods of applying protective coating such as by painting or galvanizing, although of undoubted use, have never proved entirely satisfactory and at best are only palliatives. The use of materials which are in themselves non-corrodible, obviously holds out much more promise.

The use of chromium steel in this direction is not the result of the efforts of any one individual, but rather of developments dating back many years, and in which many investigators have taken part.

The history of chromium steel is, in fact, of such interest that the author believes the space devoted to a recapitulation here of its main features is well worth while. Further, although nickel is used in conjunction with chromium in the more recent types of non-corrodible steels, the use of nickel steels without the addition of chromium has not been very successful, that is, in combating corrosion. Chromium must, therefore, be regarded as the true basis upon which the present development in the direction of non-corrodible steels has been built.

In reading this short historical review it should be borne in mind that development of the use of chromium steel of lower percentages was for many years in directions other than for resisting corrosion, chromium steel having been proved of great value for many other engineering purposes.

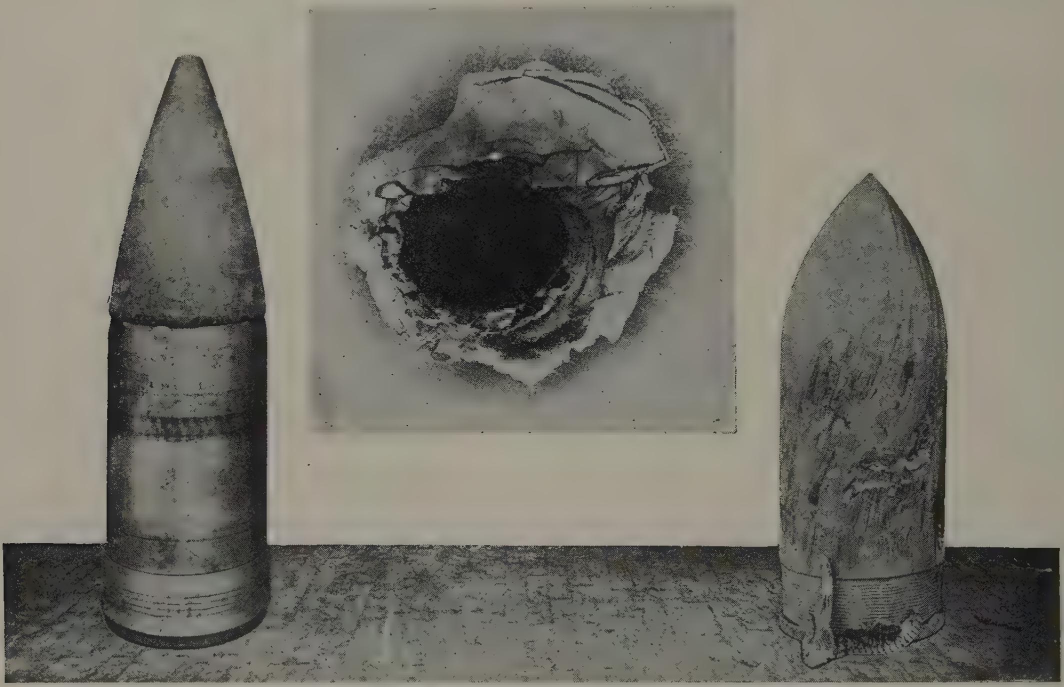

FIG. 4. ARMOUR-PIERCING PROJECTILE OF 16 IN. CALIBRE

Before and after successfully perforating a hard faced armour plate of modern type, under specially difficult conditions of attack

Reference to such developments and to those who have taken part in them, are necessarily subordinated here, although it is obvious that being the prime cause of the increasing use of chromium steel their effect has been considered when studying the development of its use for corrosion resisting purposes by the increased opportunities which it afforded for observation of, and experiment upon, its characteristics in this direction.

The great French chemist, Louis Nicolas Vauquelin, discovered chromium in 1798. This followed an investigation made by him on the "plomb rouge de Sibérie," a native chromate of lead from that country. This "plomb rouge" is now known as Vauquelinite. Vauquelin, who passed through the stormy days of those stirring times, shortly afterwards presented a communication as a Member of the Académie des Sciences, of which body he became President in 1819.

Berthier made a considerable number of experiments with the metal discovered by Vauquelin, and described these in his paper, "Sur les alliages du chrome avec le fer et avec l'acier," which appeared in *Annales des Chimie*, Vol. XVII, 1821.

It is interesting to recall that Faraday included low chromium steel in his experiments with a view to developing an untarnishable metallic mirror, although the particular experiments were not successful. Berthier frankly stated that the idea of introducing chromium into steel had been suggested to him by Faraday's interesting paper on "Alloying Different Metals with Steel." This is important, as it has been stated that Faraday obtained his idea from Berthier's work, which was not the case. Nevertheless, Berthier's experiments were much more exhaustive, and his paper referred to, contains probably the earliest published record of the acid resisting character of alloys of iron with chromium.

Later, in 1865, Julius Baur, of New York, carried out some experiments, and took out patents for combining chromium directly with iron or steel by means of ore additions. His process was worked out by the Chrome Steel Company, of Brooklyn, New York, who showed numerous specimens of this steel at the Centennial Exhibition in Philadelphia in 1876.

The modern development on scientific lines is probably due chiefly to the work in 1876 of the late M. Henri Brustlein, of the famous French firm of Messrs. Holtzer & Co., Unieux, who made a fine exhibit at the Paris Exhibition in 1889. Brustlein also communicated a valuable paper to the Iron and Steel Institute in 1886, entitled, "Chrome Pig Iron and Steel," also an important paper to the International Congress of Mines and Metallurgy in 1889 on the subject of ferro-chromium.

In 1878, Boussingault gave some interesting information on the subject in a paper which appeared in *Annales des Chimie*, Vol. XV, page 121, and noted the difficulty of dissolving iron chromium alloys for the purpose of analysis in nitric acid and even in aqua regia.

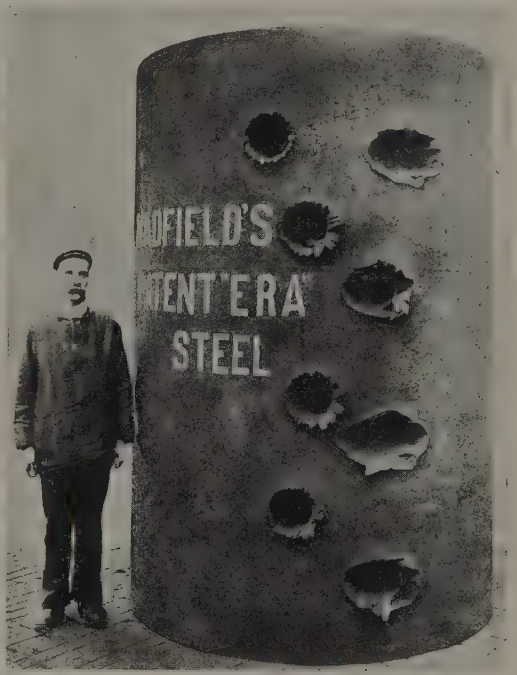

Fig. 5. Cast Ammunition Tube of Special Armour Steel

Gruner contributed a paper in 1883 to *Annales des Mines*, in which he concluded that the presence of small percentages of chromium assisted rather than resisted corrosion in moist air, sea water, and acidulated water.

MM. Carnot and Goutal in a paper published in *Comptes Rendus*, Vol. CXXVI, 1898, recorded the difficulty of dissolving iron chromium alloys, and established the important fact of the influence of carbon in such alloys. They stated that chromium steels containing a small percentage of chromium are readily attacked even by cold dilute acids when containing much carbon, but very slowly when containing a small proportion of carbon.

In 1892, the author read a paper before the Iron and Steel Institute describing his researches on chromium steel. In a report by the late Professor Floris Osmond accompanying that paper recording his examination of the author's alloys, he specially called attention to the remarkable resistance to acid attack which he had experienced with certain of the specimens, specially the one containing about 9 per cent of chromium, in endeavouring to etch them for micrographic examination.

Two of the alloys which the author produced, known as Nos. 11/L and 12/M, showed the following compositions—

C	Si	S	P	Mn	Cr
·71	·36	·10	·03	·25	9·18
1·27	·38	·10	·03	·25	11·13

It will be seen that these come within the range of some of the rustless chromium steels employed to-day. The carbon is somewhat high in percentage, due to the raw ferro-chromium of that period containing high carbon. On the other hand, a considerable quantity of steel for cutlery purposes made to-day in America contains quite as much carbon as specimen No. 11/L, also about 1½ to 2 per cent silicon. These have been fully described by Armstrong.

The patent specification of Emile Placet in 1896 discloses his appreciation, even at that time, of many benefits, besides that of reduced corrodibility, conferred on other metals by the addition of chromium.

In his British specification No. 202, which is more specially concerned with means of introducing chromium into alloys so as to obtain these benefits, he states that chromium "improves all metals and alloys with which it is mixed, by imparting to them the qualities peculiar to itself. It renders them harder, more resistant to shocks, tension, and friction, and also renders them more proof against the destructive action of the air, moisture, acids, and high temperatures." Other advantages mentioned are, in certain cases, increased electrical resistance, anti-magnetic properties, and sonorousness.

An interesting feature of this patent is the emphasis placed on the necessity for purity in the chromium used. Placet points out that from earlier experiments in this direction "no practical results could be obtained, because the chromium hitherto obtained was not pure metallic chromium, but only a more or less impure carburet of chromium.

"It was found impossible, in fact, to get rid of the carbon and other impurities contained in the chromium carburet, and consequently, when attempts were made to produce alloys with this chromium carburet, these alloys were found to be impaired by the presence of carbon and other impurities which the chromium carburet contain." His more successful results were only achieved by the use of chromium prepared by electrolysis, and which he describes as "absolutely pure," which had then become obtainable.

A valuable paper by Dr. Monnartz appeared in the German publication *Metallurgie*, of March, 1911, describing researches, specially in regard to their resistance to acid corrosion, on alloys of iron and chromium of very low carbon content, also nickel-chromium alloys. The paper also contains interesting historical references to which the author is indebted for certain of the information now given. This is an exceedingly important paper, and much of the work of the last twenty years relating to the subject, appears to have proceeded from it.

Useful researches were also carried out by Messrs. Strauss and Maurer on non-corrodible chromium steel containing 20 per cent of chromium. These investigations were referred to in a paper by Daeves which appeared in *Stahl und Eisen*, of 24th August, 1922, and in which reference was made to high chromium alloys. The experiments are said to have been carried out in 1909 and 1910, but as far as is known the results were not published at the time. Numerous acid tests were carried out showing the great resistance of these products.

From 1912 to 1914, Mr. Harry Brearley carried out his well-known experimental work on high chromium steel, arising in connection with a search for an improved material for rifle barrels and liners for gun tubes. It is understood that although the experiments to discover a steel highly resistant to gaseous erosion did not lead to any immediate practical results, and that the work was not originally inspired by any intention or hope of discovering stainless steel, observations made at the time of the experimental work referred to led to the discovery by Mr. Brearley that by hardening, tempering, and polishing, steel of such character could be made into stainless cutlery. This was followed by numerous developments in the application of chromium steel by many others in this country, the United States, France, Germany, and elsewhere. An example of one such application is illustrated in Fig. 6. The list of these will no doubt be increased as experience is acquired, possessing as this material does, besides a high resistance to corrosion, excellent strength and ductility.

The author, when presiding at the Symposium on the "Corrosion of Metals; Ferrous and Non-Ferrous," held by the Faraday Society in December, 1915, that is, during the war, in his introductory address, entitled "The Corrosion of Steel Alloys," specially referred to the important work done on the alloy known as stainless steel, which had then recently come before the metallurgical world. He pointed out that much credit was due to Messrs. Thomas Firth & Sons, Mr. Harry Brearley, and others, for the work they had done in this matter.

It should also be mentioned that although chromium has proved of such value as an alloying element in this connection, alloys of iron and silicon containing 15 to 20 per cent of the latter element, also offer considerable resistance to the attack of certain chemical agents, notably the mineral acids, and mixtures of these. For this reason, and from the fact that excellent castings can be made in such material, fairly considerable use has been made of it in the form of vessels and pipes used in chemical manufacturing plants. Its brittleness, however, greatly detracts from its more extended application.

A small addition of copper to steel has also been found beneficial in certain limited directions, notably in resisting the specially corrosive action of certain industrial atmospheres.

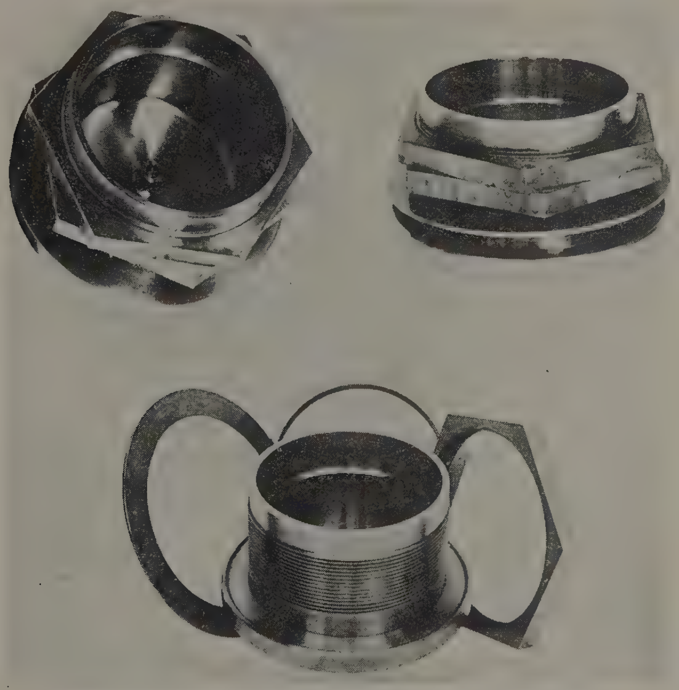

Fig. 6. Hydrophone Diaphragms, for Submarine Signalling, of Non-rusting Chromium Steel

The American, Mr. Elwood Haynes, also claimed to have done much work in this direction, but at any rate his early work was more associated with alloys of chromium with cobalt. A paper by him on this subject was read before the American Iron and Steel Institute in May, 1921.

Much useful and interesting information on this whole subject is also given in the recently published German book, *Die Edelstähle*, by Dr-Ing. F. Rapatz.

It is further interesting to refer to the material known as Ingot Iron—really a steel in the method of its manufacture—and in which a high degree of purity is aimed for, that is freedom from all elements other than the iron itself, so far as this can be attained. To the more usual forms of rusting, this material has proved distinctly more resistant than ordinary iron and steel. It has consequently achieved very considerable use in many directions. Thus, in this instance, some success against the activities of corrosion has been attained not by alloying but rather the opposite.

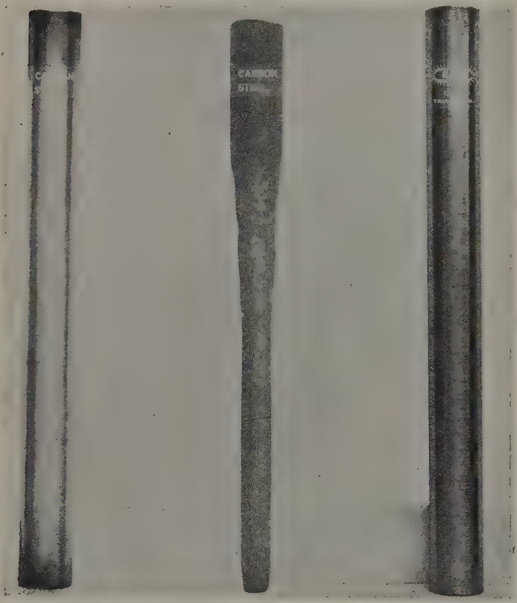
Ordinary carbon steel (untested)
Ordinary carbon steel, 45 per cent loss in weight by corrosion
Non-corrodible steel, no loss

FIG. 7. TESTED SPECIMENS

Illustrating the resistance to corrosion of a non-corrodible steel of modern type, after 24 hours' immersion in nitric acid

To return to non-corroding or stainless steel, this was, however, only a partial solution of the corrosion problem. While resistant to certain forms of corrosion, it is liable to be badly attacked by others. Further, to enable it to display its qualities to the fullest extent, heat treatment and the preparation of a smooth surface are necessary, special care being required in both respects otherwise failure may occur.

With such progress to build upon, however, further research has led into a fruitful field among the nickel-chromium steels which have the desired qualities to a much enhanced degree.

One of the earliest investigators with regard to nickel-chromium alloys was the French firm known as La Société Anonyme La Neo-Metallurgie, who, in 1902, pointed out that alloys of iron, nickel, and chromium presented great resistance to corrosion. The exact words used by this company, describing the qualities of their materials, are as follows—

" In these conditions of purity, the elements, nickel and chromium, modify very advantageously the properties of the products, steels and irons, in which they are incorporated. In tractive tests, great increase in the resistance of these substances to rupture is found. Their limit of elasticity is also largely increased without their extension experiencing a reduction of any importance. But the most important modification to which these elements, nickel and chromium, give rise, consists in the important property (which forms the subject of the researches of metallurgists at the present time) which they communicate to the said products, steels and irons, of presenting great resistance to corrosion."

It is interesting to note that the specification of the British patent No. 23,861, of 1903, shows that the discovery of these useful properties concerned materials in which iron is alloyed with high percentages of nickel and chromium ; in the patent, nickel is stated to vary in certain types, between 5 and 60 per cent, the chromium from about 24 to 57 per cent. It is also further specified that carbon should be in a very small proportion, and the alloy as a whole of a high degree of purity.

In 1917, La Société Anonyme de Commentry Fourchambault et Decazeville, Imphy, France, in whose laboratories were carried out the researches of the well-known metallurgists, Dumas, Guillaume, Chevenard, and others, had produced their high chromium-nickel steels, which have proved to possess specially valuable properties in various applications. These have subsequently been developed in this country in collaboration with Hadfields, Ltd., of Sheffield, and have resulted in the manufacture of a range of alloy steels possessing not only high resistance to various types of corrosion, but also to oxidation and the effects of high temperatures.

(*e*) **Steel for Service at High Temperatures.** Advances in engineering practice are more and more calling for materials having the virtues of steel, but at the same time able to be employed at high working temperatures even up to well above a red heat. Valves of internal combustion motors illustrate a simple instance, while the introduction of mechanical appliances into the heating space of furnaces, as, for example, mechanical stokers and automatic reheating furnaces, provide many such cases. The future of the internal combustion turbine, and advances in existing internal combustion engines are also largely wrapped up in the question of obtaining suitable materials.

Provided temperatures are not excessive, ordinary steels or cast iron, or simple alloy steels often meet requirements, but as is well known, at temperatures above a red heat these lose most of their strength and become incapable of sustaining continuously even their own weight. If the atmosphere is at all oxidizing, these too waste away rapidly by "scaling."

It is gratifying, therefore, to find that metallurgical research, again through the medium of alloy or steels, is able to keep abreast of such requirements. It is now possible to provide special steels which possess considerable rigidity under these conditions, and "Era" heat resisting steel at 700° C. may be loaded up to a tensile stress of 31 tons per sq. in., that is, it has at this temperature a tenacity similar to that of mild steel at atmospheric temperature. Under continuous loading, this special steel will sustain a stress of 6·5 tons per sq. in. without suffering deformation however long the stress is maintained, that is, beyond a slight initial creep. This material also does not become covered with scale at these high temperatures, nor is it attacked by furnace gases, even those containing large quantities of sulphurous oxide. In addition, it possesses a marked resistance to corrosion at ordinary temperatures. This special steel is being used for many purposes where duty at high temperatures, up to 1,000° C. and over is required, such as for the construction of parts to work in the interior of furnaces, also case-hardening boxes, pyrometer tubes, recuperator tubes, and many other applications.

Castings of the same steel are used with considerable success for certain parts known as slicing links in the chain grates of the boiler plant of a large power station in the North of England. These links work in what is probably the hottest part of the fire using a draught of preheated air, and are subject to considerable abrasion. Yet they have an average life of over 1,000 hours actual steaming time. Links of ordinary cast steel, even when partially protected by a special surface coating, lasted only six days. This particular application of "Era/H.R." steel thus provides a further example of the true economy, previously referred to, which can result from the use of alloy steels.

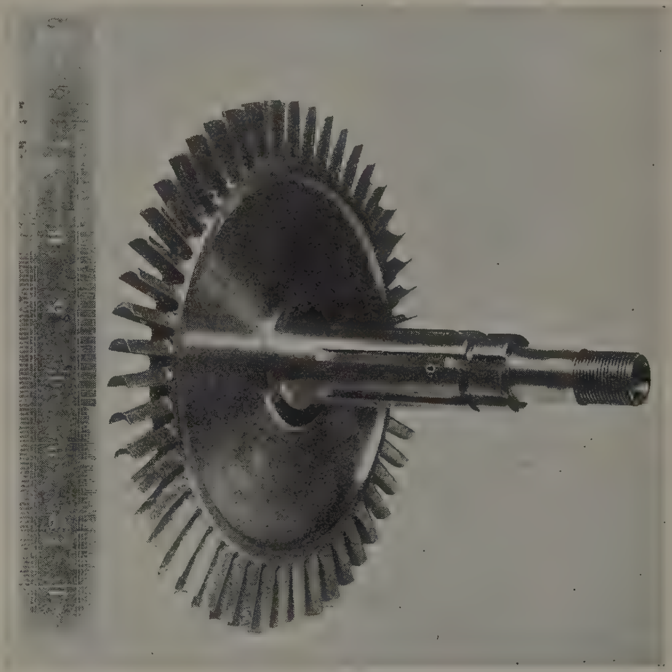

Fig. 8. Exhaust Turbine Rotor of ERA/A.T.V. Heat-resisting Steel

An outstanding application of a further type of heat resisting steel, known as "Era/A.T.V.," is shown in Fig. 8. These rotors are driven by the exhaust gases from internal combustion engines, and work continuously at a temperature from of 800 to 950° C., also run up to the very high testing speed of 53,000 revolutions, and working speed of about 30,000 revolutions per minute.

The successful operation of these rotors at a bright red heat, and under exceedingly high centrifugal and other stresses, at the same time being subjected to the corrosive and erosive effect of the hot gases, affords a wonderful example of the advances made in this field.

(*f*) **Other Alloy Steels.** Other alloy steels, of which only brief mention can be made here, but which nevertheless help to illustrate the remarkably diverse and useful properties obtainable by alloying, are the following—

High speed steel, an alloy containing about 16 per cent of tungsten and 4 per cent of chromium, and carbon to the extent of about 0·6 per cent.

This steel, like carbon steel, is hardened by rapid (or even comparatively slow) cooling from a high temperature, and so is suitable for tools for cutting metals. In addition, however, it retains its hardness and cutting powers under the high temperatures—even at a low red heat—developed in the tool when heavy and rapid cuts are being taken, and under which conditions carbon tool steel becomes softened by tempering. High speed steel, through the improvements it has enabled to be effected in machining practice, has been of inestimable value. In the latest types about 1 per cent of vanadium has been added, to the enhancement of its qualities, while other elements, such as cobalt and molybdenum have been tried with some success.

Invar, an alloy of iron with 36 per cent of nickel, which does not expand or contract with change of temperature, is useful therefore for many purposes, such as surveyors' measures, and pendulums, and balances of clocks and chronometers. A great work has been done in this direction by M. Guillaume, head of the famous laboratories at Sèvres.

Elinvar is an alloy of the same class whose modulus of rigidity does not vary with temperature, providing springs and tuning forks of unvarying characteristics. Both these steels have been introduced by La Société Anonyme de Commentry Fourchambault et Decazeville.

Permanent magnets, containing about 6 per cent of tungsten and 0·6 per cent of carbon, were a great improvement in respect of strength and permanence over magnets of carbon steel.

More recently the use of a steel containing about 33 per cent of cobalt has effected a still greater improvement, raising the coercive or demagnetizing force from about 70 to over 200. This discovery is stated to be due to Professor K. Honda of Japan. Hardening elements, such as chromium and tungsten, with carbon, are also necessary in this case.

In the steel known as "A.M.F." has been provided a material suitable for working parts of plants for the production of low temperatures, such as are used in the liquefaction of gases. While most ferrous materials, otherwise tough, become extremely brittle and, therefore, unsuitable under these conditions, "A.M.F." preserves its toughness at low temperatures.

CONCLUSION

In the foregoing by no means exhaustive description of some of the applications of special steels, it is hoped sufficient has been said to show the very great importance of the subject, the wide range of application, and the tremendous possibilities for research in this field. With regard to the future, in spite of the very large amount of study the subject has received, it has not yet been found possible to arrive at general principles of such wide validity that useful new alloy steels may be predicted by deduction.

It is by means of carefully planned series of experiments, accurate observation and cautious judgment in drawing conclusions, that the valuable special steels already known have been discovered. Further, it should not be imagined that the commercial production of a new alloy is possible without long and sustained investigation on an increasing scale of magnitude and expense, following the original discovery. The properties which render it valuable are frequently such as to occasion special difficulties in manufacture, heat treatment, and even in methods of control. It often happens also that a considerable time elapses before the possibilities of practical application of a particular alloy are realized by those who ultimately benefit by its use, and the introduction of a new material may involve the displacement of costly plant, and present fresh problems of design and construction.

Difficulties of the kind indicated beset the introduction by the author of both manganese and silicon steels, and only by long continued perseverance was he able to overcome them. There is, therefore, no royal road, and it will probably be found that future progress in this direction will have to be achieved by a continuance of the patient and well considered labours of those who have devoted themselves to its pursuit.

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XV

STRUTS

Struts may be defined as members which are subjected to end compressive loads and, in accordance with the ratio

$$\frac{\text{length of strut}}{\text{least lateral dimension of strut}}$$

so the strut is classed as *short*, *medium*, or *long*. The lateral dimension usually taken is the *radius of gyration* of the cross-section, and so, the above ratio becomes

$$\frac{\text{length of strut}}{\text{least radius of gyration of cross-section}}$$

$$\text{written } \frac{l}{k}$$

There are no rigidly fixed lines of demarcation between *short*, *medium*, and *long* struts, but it is usual to regard a *long* strut as one in which $\frac{l}{k}$ exceeds, say, 150, and a *short* strut as one in which $\frac{l}{k}$ is less than, say, 40.

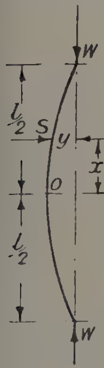

Fig. 75

Struts are of such frequent occurrence in engineering practice that a reliable theory of their strength is greatly to be desired. Euler has investigated the strength of long struts when loaded under ideal conditions, and we shall give his theory later, but we may note here that, unfortunately, the actual conditions of loading and supporting or fixing are often far different from the ideal. This has led many experimenters to frame *empirical formulae*, and of these, the simplest and most satisfactory is probably the Rankine-Gordon formula.

Euler's Strut Formula

Euler's theory is based on certain *ideal conditions*: (1) The line of action of the compressive load must pass through the $C.G.$ of every cross-section; (2) the strut must be straight and of constant cross-section; (3) the weight of the strut itself is neglected; and (4) the material of the strut must be homogeneous and of uniform elasticity. Assuming the ideal conditions given above, we may now proceed to deduce

Euler's Theory of Long Columns

Consider a strut of length "l" acted upon by an end compressive load W, which bends the strut into the shape shown in Fig. 75, the ends being secured by frictionless pin joints. At any section "S," the bending moment is Wy, and since the deflection is due to bending as a beam, we may use the approximate expression

$$\text{B.M.} = Wy = -EI\frac{d^2y}{dx^2}$$

$$\text{or } \frac{Wy}{EI} = -\frac{d^2y}{dx^2} \quad . \quad . \quad . \quad . \quad (1)$$

The solution of equation (1) is

$$y = A\sin\sqrt{\frac{W}{EI}}\times x + B\cos\sqrt{\frac{W}{EI}}\times x \; . \; (2)$$

We have now to evaluate the constants A and B.

Differentiating (2), we get

$$\frac{dy}{dx} = A\sqrt{\frac{W}{EI}}\cos\sqrt{\frac{W}{EI}}\times x - B\sqrt{\frac{W}{EI}}\sin\sqrt{\frac{W}{EI}}\times x \quad . \quad . \quad . \quad (3)$$

$$\text{Now, at the ends where } x = \frac{l}{2}, y = 0 \quad . \quad (4)$$

and also, at the centre, where $x = 0$

$$\frac{dy}{dx} = 0 \quad . \quad . \quad . \quad . \quad . \quad (5)$$

Inserting these relations, (4) and (5), in (2) and (3) respectively, we get

$$0 = A\sin\sqrt{\frac{W}{EI}}\times\frac{l}{2} + B\cos\sqrt{\frac{W}{EI}}\times\frac{l}{2} \; . \; (6)$$

and

$$0 = A\sqrt{\frac{W}{EI}}\cos\left(\sqrt{\frac{W}{EI}}\times 0\right) - B\sqrt{\frac{W}{EI}}\sin\left(\sqrt{\frac{W}{EI}}\times 0\right) \quad . \quad (7)$$

$$\text{or } 0 = A\sqrt{\frac{W}{EI}} - 0$$

But since $\sqrt{\frac{W}{EI}}$ cannot $= 0$, $\therefore$ A must $= 0$.

Equation (2) can now be written as

$$y = B \cos\left(\sqrt{\frac{W}{EI}} \times x\right) \qquad (8)$$

and when $x = 0$, this becomes

$$y = B \cos 0, \text{ or } y = B.$$

Again, $y = 0$ when $x = \pm \frac{l}{2}$; substituting $+ \frac{l}{2}$ in (8), we get

$$0 = B \cos\left(\sqrt{\frac{W}{EI}} \times \frac{l}{2}\right) \qquad (9)$$

Now $\sqrt{\frac{W}{EI}} \times \frac{l}{2}$ is the radian measure of an angle whose cosine value is 0, and since we require the *least* load which will cause the strut to fail we must find the *least* value of $\sqrt{\frac{W}{EI}} \times \frac{l}{2}$ this is clearly $\frac{\pi}{2}$, hence we may write

$$\sqrt{\frac{W}{EI}} \times \frac{l}{2} = \frac{\pi}{2}$$

or

$$W = \frac{EI\pi^2}{l^2} \qquad (10)$$

Equation (10) is Euler's expression relating the *buckling load* " W " with the *length of the strut* " l," the *moment of inertia* of the cross-section of the strut " I," and the *modulus* of *direct elasticity* " E " of the material of which the strut is made, the strut having frictionless pin-jointed ends. This, together with the other cases which differ from the above in the method of supporting or fixing the ends, are given in the table below—

Method of supporting or fixing the ends of the strut	Euler's buckling load " W "
One end fixed and one end free	$W = \frac{EI\pi^2}{4l^2}$
Both ends pin jointed	$W = \frac{EI\pi^2}{l^2}$
One end fixed and one end pin jointed but guided in endwise movement	$W = \frac{2{\cdot}04EI\pi^2}{l^2}$
Both ends rigidly fixed	$W = \frac{4EI\pi^2}{l^2}$

Medium Length Struts

Rankine-Gordon Formula. The simple expression suitable for a short strut is

$$W_1 = f_c \times A$$

$$\text{or} \quad \frac{1}{W_1} = \frac{1}{f_c \times A}$$

where f_c = compressive stress,
and A = area of cross-section.

The strength of very long struts is given with considerable accuracy by Euler's expressions, which are of the form

$$W_2 = c \times \frac{EI\pi^2}{l^2}$$

$$\text{or} \quad \frac{1}{W_2} = \frac{l^2}{cEI\pi^2}$$

where c = a constant
E = modulus of direct elasticity
I = moment of inertia of cross-section
l = length of strut.

If we write $\frac{1}{W_3} = \frac{1}{W_2} + \frac{1}{W_1}$ . . (11)

where W_3 = buckling load for strut of *any* length, we have an expression suitable for both *short* and *long struts*.

By substituting into (11), we get

$$\frac{1}{W_3} = \frac{l^2}{cEI\pi^2} + \frac{1}{f_c \times A} \qquad (12)$$

If the stunt is *short*, l will be a small quantity, and the fraction $\frac{l_2}{cEI\pi^2}$ will be negligible by comparison with $\frac{1}{f_c \times A}$ (in which *length* does not enter), and the expression (12) can now be written

$$\frac{1}{W_3} = \frac{1}{f_c \times A} \qquad (13)$$

that is, (12) reduces to the expression for the *short* strut.

Again, if the strut is *long*, that is, " l " is great, then l^2 is much greater, and the term $\frac{1}{f_c \times A}$ is negligible by comparison with $\frac{l^2}{cEI\pi^2}$. The expression (12) now reduces to

$$\frac{1}{W_3} = \frac{l^2}{cEI\pi^2} \qquad (14)$$

which is that for a long strut. We have now shown that (12) is applicable to both *long* and *short* struts, and we may reasonably assume the expression to hold for *medium* length struts as well.

Equation (11) is $\frac{1}{W_3} = \frac{1}{W_2} + \frac{1}{W_1}$

or $$\frac{1}{W_3} = \frac{W_1 + W_2}{W_1 W_2}$$

and inverting, we get
$$W_3 = \frac{W_1 W_2}{W_1 + W_2}$$
dividing numerator and denominator by W_2, we get
$$W_3 = \frac{W_1}{1 + \frac{W_1}{W_2}}$$
and now substituting for W_1 and W_2
$$\therefore \qquad W_3 = \frac{f_c A}{1 + \frac{f_c A l^2}{c E I \pi^2}}$$
and since $I = Ak^2$
$$\therefore \qquad W_3 = \frac{f_c A}{1 + \frac{f_c A l^2}{c E \pi^2 A k^2}}$$
and since f_c, A, c, E, and π^2 are constants, we may write
$$W_3 = \frac{f_c A}{1 + a\left(\frac{l}{k}\right)^2} \qquad (15)$$
the form of the Rankine-Gordon formula.

The value of the constant "a" in (15) depends upon the end fixing conditions, whilst f_c depends upon the material as tabulated below.

An example will now be given to show the application of the Rankine-Gordon formula.

EXAMPLE. A stanchion 36 ft. high has its ends rigidly fixed. The stanchion consists of a R.S.J., 15 in. × 6 in., web ½ in. thick and flanges 0·88 in. thick. To each flange are riveted two plates, 12 in. × ⅝ in. Calculate the safe axial load by means of the Rankine-Gordon formula, the factor of safety being 5.

We have to determine the least moment of inertia of this section, and this will clearly be about the axis BB (Fig. 76).

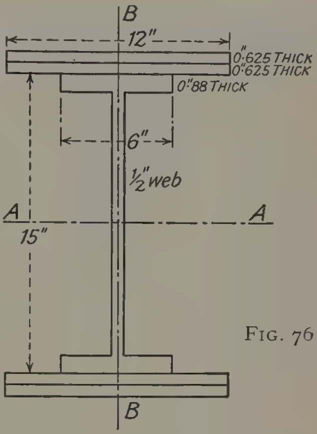

FIG. 76

$$I_{BB} = \Sigma \tfrac{1}{3} bd^3$$
$$,, \quad = \tfrac{1}{3}(2\cdot5 \times (6)^3 + 1\cdot76 \times (3)^3 + 13\cdot24 \times (\tfrac{1}{4})^3) \times 2.$$
$$,, \quad = \tfrac{2}{3}(540 + 47\cdot52 + 0\cdot207) = 391\cdot8 \text{ inch}^4 \text{ units.}$$

Area of Cross-section

$$A = 30 + \tfrac{1}{2} \times 13\cdot24 + 6 \times 1\cdot76 = 47\cdot18 \text{ sq. in.}$$

Also $I = Ak^2 \quad \therefore \quad k^2 = \frac{I}{A} = \frac{391\cdot8}{47\cdot18}$

Substituting these values in equation (15).
$$W_3 = \frac{f_c A}{1 + a\left(\frac{l}{k}\right)^2}$$
$$\text{or } W_3 = \frac{21 \times 47\cdot18}{1 + \left[\frac{1}{30,000} \times \frac{432 \times 432 \times 47\cdot18}{391\cdot8}\right]}$$
$W_3 = 566\cdot3$ tons buckling load.

The factor of safety being 5, the safe working load $= \frac{566\cdot3}{5} = 113\cdot26$ tons.

Material	f_1 tons per sq.in.	Values of a		
		Fixed ends	Pinned ends	One end pinned One end fixed
Cast iron . .	36	$\frac{1}{6400}$	$4 \times \frac{1}{6400} = \frac{1}{1600}$	$2 \times \frac{1}{6400} = \frac{1}{3200}$
Mild steel . .	21	$\frac{1}{30,000}$	$4 \times \frac{1}{30,000} = \frac{1}{7500}$	$2 \times \frac{1}{30\ 000} = \frac{1}{15,000}$
Wrought iron . .	16	$\frac{1}{36,000}$	$4 \times \frac{1}{36,000} = \frac{1}{9000}$	$2 \times \frac{1}{36,000} = \frac{1}{18,000}$

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.[1]

LESSON XIII

THE RECIPROCATING MACHINES

This group, which is characterized by the fact that for each cutting stroke, there is a lost return stroke, includes the planing, shaping, slotting, and broaching machines. The planers are more than 100 years old, the shaper was invented by Nasmyth, the slotter came a few years later, and the broaching machine is recent. The first three long held a leading position in the shops until their monopoly was attacked by the machines using rotary cutters. Under the stress of competition, they have taken a new lease of life. But the machines of the present day differ in every detail from their early predecessors—in their methods of driving, and in much higher efficiencies, so that their position, once jeopardized, is now firmly re-established.

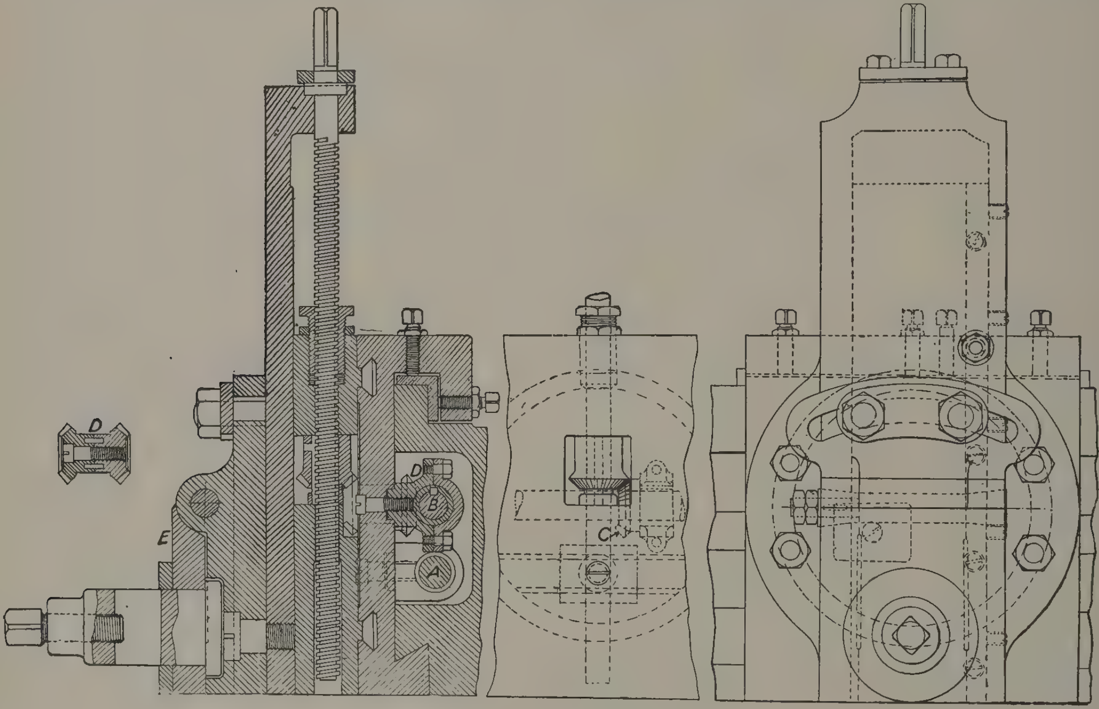

Fig. 90. Tool Box of Planer

The Planing Machines

The General Design. This includes a massive work table reciprocating on a deep bed, flanked with upright housings tied at the top, and receiving on their faces a cross rail that is adjustable vertically, and along which the tool boxes have feeding movements, transverse and vertically. Many machines also have a tool box on one or on both housings with vertical feeding. The capacity of a machine is that of the dimensions of the work which it can handle. It includes the length of travel of the table, the width between the housings, and the maximum height from the work table to the lower side of the cross rail. For work that exceeds these limitations, open-side machines, and side planers are built, and the vertical-cutting machines, the wall creepers. The planing machine has thus many variants.

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[1] We regret to announce that Mr. Joseph G. Horner died on 9th February at his residence at Bath, after a brief illness.

The Movements of the Planer. In the common design, these are the travel of the table that carries the work past the tools, and that of the tool boxes fed along the cross rail, with provision for feeding downwards perpendicularly, or at an angle. The cross rail that carries the tool boxes is adjusted vertically on the housings with screws and bevel wheels acting in unison. The table slides on vee'd or on flat ways, and is lubricated with rollers that float in oil pockets. Its range of movements for work of various lengths is limited and controlled, and its reversals are effected in all the old designs with dogs, which are set and clamped in a groove that extends along one edge, and which, striking tappets, actuate the reversing mechanism. At the same time the feed of the tool boxes is put on. These are fed along the cross rail with a screw *A*, Fig. 90, at various rates, duplicated when there are two tool boxes. The down-feed of the tool slide is derived from a splined shaft *B*, that lies parallel with the screw within the cross rail. A bevel gear *C* splined to the shaft and travelling with the box, feeds, through a pair of bevels *D*, the screw of the tool box at a rate controlled by a ratchet or feed disc at one end of the rod. On the face of the saddle an intermediate plate can be swivelled with an annular fitting to any angle. This receives the vertical slide that carries the tool holder or clapper *E*, hinged to it to relieve the pressure on the point of the tool during the return stroke. The clapper plate can be set at an angle to give clearances when setting tools at an angle, and relieve the tool on the return stroke. A single tool is used in a holder. If two are required, two boxes are mounted on the cross rails.

Double Cutting. The objection to the lost stroke has been the cause of designs in which tools cut on both strokes. The earliest was the Whitworth rotating holder, termed the "Jim Crow," because it turned sharply round 180° after each cut. It may be retained yet, but other designs have been schemed, in which a tool box carries two tools set back to back, and tilts, so that when one is cutting the other is lifted clear off the work. These are used only to a moderate extent. Fittings and adjustments must be very precise to enable one tool to follow in the track of another, and produce accurate results. Sometimes a roughing and a finishing tool are mounted side by side, one to follow close behind the other. But efficiency is generally sought by employing one tool for roughing, and another for finishing, with a broader feed, by speeding up both for cutting and return, and providing variations in speeds for different materials, and for roughing and finishing.

The reason why additional tool boxes are fitted on one or on both housings is partly to increase the capacity of the machine, but chiefly to permit of tooling deep angular and vertical faces. These can only be dealt with from the cross rail by using tools that overhang much from their holders, with resulting vibration and chatter. The tools on the side heads can be brought near to the work with little overhang, if the face to be planed is brought close to the edge of the table.

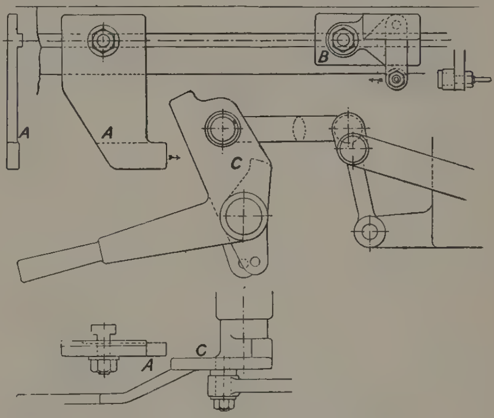

FIG. 91. DOGS AND TAPPET OF PLANER

Methods of Driving and Reversal. Space will not permit of a description of the numerous methods that have been employed for driving the planers. They were long confined to the very bad method of switching a single belt from a loose central pulley over flanking pulleys, the driving and reverse being effected with noisy gears within the bed, terminating with a small pinion driving the table rack. This is wholly abandoned in favour of pairs of fast and loose pulleys, larger for driving and smaller for reverse, with open and crossed belts, with a large wheel meshing with the table rack, or with a spiral drive. In many of the later designs an all-electric drive is introduced.

The planers have been improved by the inclusion of higher and variable cutting speeds, in higher rates of return, and in regenerative cushioning arrangements, to give back energy to the table on reversal. At one time much credit was taken for the quick return during the non-cutting stroke. But it was only twice or thrice the speed of cutting, and the latter was slow, seldom exceeding 20 ft. per minute, and this for all materials, all depths of cut and widths of feed. Then, for awhile, emphasis was laid on the quickened rates of return in some American machines—four, six, and even eight times, still oblivious of the more important matter of the ratio between the two. At present the " cycle," the total time of cutting and return, receives most attention, and machines are built to give a range of cutting speeds, three, or more, with fixed or variable speeds of return.

Belt Shifting. When belts are shifted between fast and loose pulleys, the method employed is not that of a direct control of the forks with a shipper bar as of old, but the movement is sharper. One example among several is shown in Figs. 91 and 92. The dogs clamped against the edge of the table at *A* and *B*, Fig. 91, set for the working stroke, strike the tappet *C* in alternate directions and shift the belt between fast and loose pulleys. In Fig. 92, an arm connected to the tappet lever imparts a linear movement to the slide *D*, sliding it under the control of two slots and studs over the casting that covers the two pairs of fast and loose pulleys for driving and reversal. The centre of the sliding plate is occupied with a cam groove of the dog-leg pattern. A boss on one side of each belt fork near the fulcrum, coerced in this by the sliding of the plate, causes the forks to be shifted over to adjacent pulleys rapidly and almost in unison.

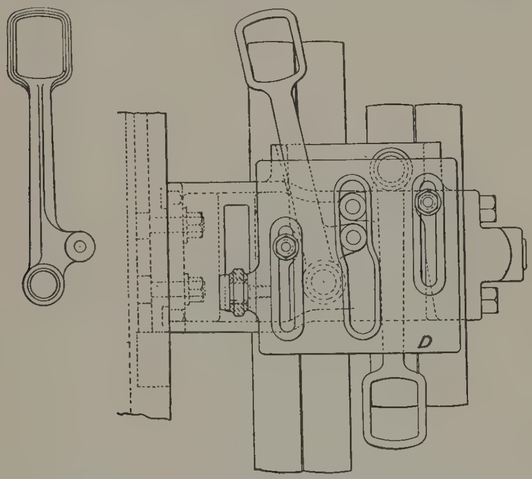

FIG. 92. CAM CONTROL OF BELT FORKS

The Reversals. The early machine makers did not know what later experiments by Captain Tresidder demonstrated, that the act of reversal absorbs about twice the amount of power that is required for the work of cutting. Attention was then directed to storing up power during cutting in a regenerative or cushioning mechanism to be given back at the moment of reverse. In the Bateman, strong coiled springs flank the table rack, one pair of which absorbs energy during their compression to restore it to the other pair when reversal occurs. The rack slides in grooves under the control of the springs, connected to it with a crosshead at each end. In another, a coiled spring surrounding the driving shaft forces a sliding clutch into engagement with a spur wheel loose on the shaft, and so drives the shaft in the return direction. When the clutches are forced apart, the spring is compressed and the spur wheel drives against the resistance of the shaft as though it were

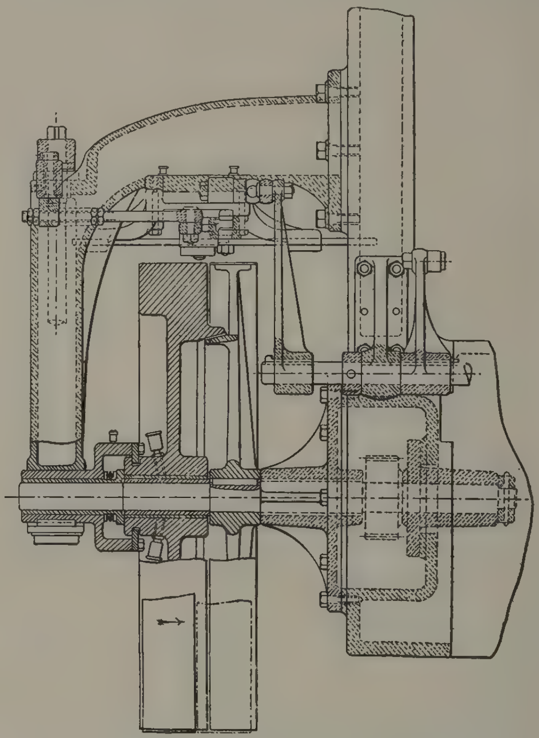

FIG. 93. PULLEY THAT STORES UP ENERGY FOR REVERSALS OF PLANER

keyed directly on it, but actually it is driven through the coiled spring. The essential feature is, that at the termination of a cutting stroke when reversal occurs, and the wheel for a moment is not driving, the spring, forcing the sliding clutch into mesh with that on the wheel drives the clutch for reverse, and acts as a buffer.

The driving pulleys now are made to assist in the reversals. They are very light—often of an aluminium alloy, because when heavy they store up energy that resists reversal. The Cincinnati Company found, by experiment, that pulleys of cast iron took 39 h.p. at reverse, while those of aluminium absorbed only 30 h.p. A light driving pulley is associated with a very heavy loose pulley in the Bateman planers, in order to store up kinetic energy to be utilized at the moment of reversal. The two are connected with a friction clutch, Fig. 93. When the belt is shifted to the fast pulley, the loose pulley is clutched to it automatically, so that both pulleys are driven by the belt, and the heavy pulley gives up its kinetic energy. When the belt is shifted to the heavy loose pulley, the clutch is shifted automatically and the pulley runs freely. The heavy pulley acts as a flywheel. It is wider than the fast one, and so the belt overlaps on the first, receiving and transmitting the energy that is stored up in it at the moment of reversal.

Feeding. Feeds are derived primarily from the reversals. The feed disc, Fig. 94, is keyed on the driving shaft. The amount of feed is regulated by the adjustment of a block across the disc, whence the drive is transmitted to a rack bar connected to the feed gears on the end of the cross rail, from which the motion of the screw for the cross traverse, and that of the splined shaft for vertical feeds are controlled. The disc slips, on reversal, through a leather washer, the degree of friction between which and the metallic faces can be regulated with screws.

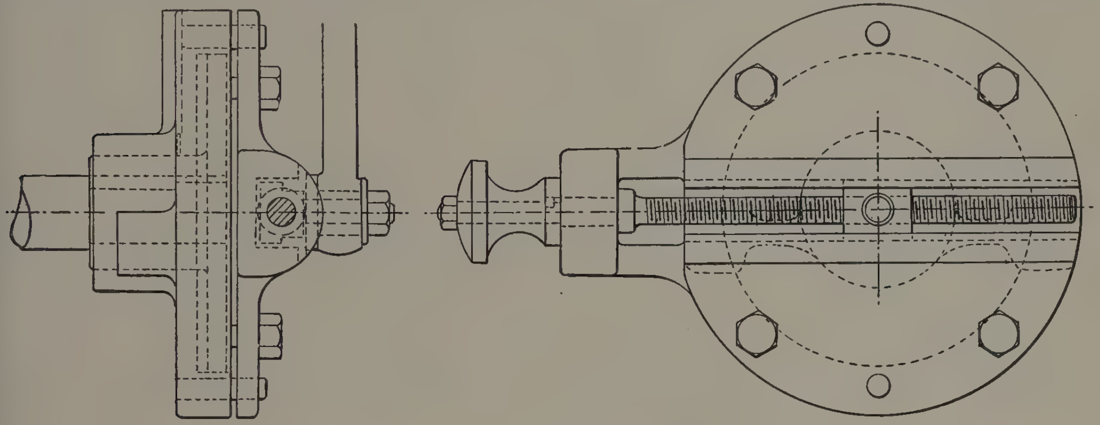
Fig. 94

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XIV

Stuffing Boxes. We made reference in the last lesson to machine details which, being common to most machines, offered for our choice a number of various existing designs. A further example is to be found in the stuffing box. This detail is required where a moving rod or spindle passes into a vessel containing a fluid or gas at a pressure other than atmospheric. Its purpose is to prevent a leakage of the cylinder contents past the rod.

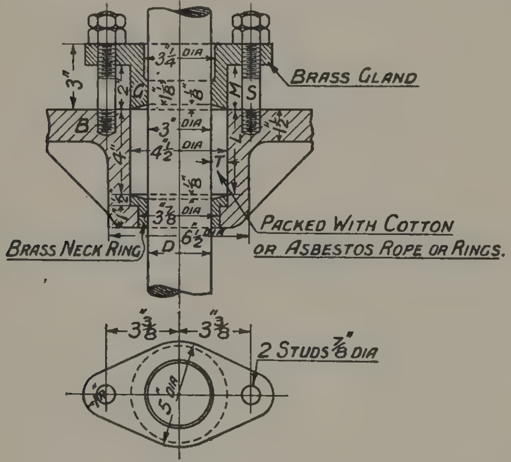

Fig. 1

A common design is shown in Fig. 1, the stuffing box *B* in this case being cast as part of the cylinder. The space between the rod and the stuffing box is packed with cotton or asbestos rope or rings, according to the nature and temperature of the cylinder contents. A gland *G* moves along the rod and has its position regulated by the gland studs *S*. It will be evident that the movement of the gland towards the cylinder decreases the packing space and so, compressing the packing, reduces the space between the rod and the packing to a minimum. In an efficient stuffing box the packing grips the rod with a force sufficiently great to avoid leakage from the cylinder, and yet not large enough to break the oil or water film lubricating the rod.

With regard to suitable proportions for this machine detail, it is surprising to find an exceptionally various practice, and the proportions given in Table I should be considered as having minimum rather than orthodox values. Moreover, it often happens that the strength of the gland demands a greater thickness than is required by the packing.

Application. Suppose we have to make a drawing of a stuffing box for a steam cylinder subject to a pressure of 200 lb. per sq. in., the rod having a diameter of 3 in. Referring to the above proportions we find the suitable packing thickness to be ⅝ in., but proceeding with the drawing we find that a space of ¾ in. is required for a neck bush of suitable dimensions. The depth of the packing space may be fixed in accordance with the accepted proportion, and we may proceed to the gland studs.

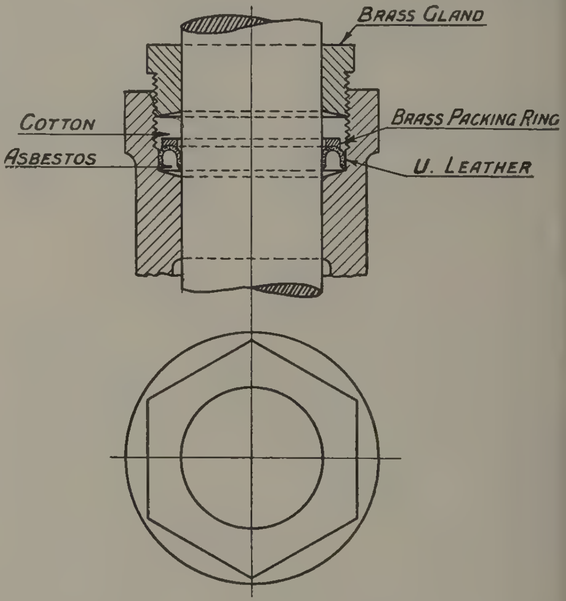

Fig. 2

In fixing the size of the studs we have to observe that the probable maximum load on the gland is determined by the size of the studs. We may say that for each size of stud there is a corresponding maximum load which the average man, using a standard spanner, can produce. Conversely, we may, considering the packing to be perfectly resilient, find the load required to produce a grip on the rod equal to the steam pressure in the cylinder.

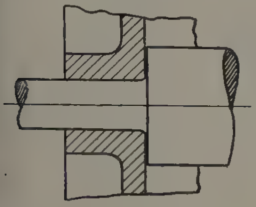

FIG. 3 (*a*)

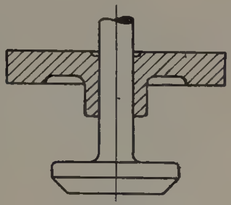

FIG. 3 (*b*)

We have, the cross-sectional area of the gland subject to pressure $= \dfrac{\pi(4{\cdot}5^2 - 3^2)}{4}$

$= 8{\cdot}82$ sq. in.

Then the gland load due to a steam pressure of 200 lb. per sq. in.

$= 8{\cdot}82 \times 200 = 1764$ lb.

Giving a 100 per cent margin over the above load to allow for the inelasticity of the packing, we find each gland stud would be subjected to a tensile load of $\dfrac{2 \times 1764}{2} = 1764$ lb.

Thus, allowing a working stress of 4,000 lb. per sq. in., the minimum area of the studs (i.e. at the bottom of the thread) $= \dfrac{1764}{4000} =$ ·441 sq. in., which corresponds approximately to the minimum area of a $\frac{7}{8}$ in. diameter Whitworth stud. We may now proceed with the drawing (referring to Table I) to the dimensions given in Fig. 1.

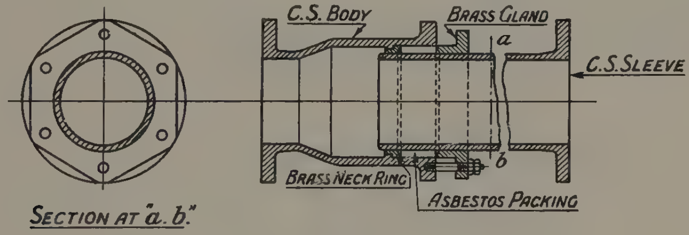

FIG. 4

Stuffing Box Omitted. The stuffing box has one feature which, when applied to lightly-loaded spindles, may prove very objectionable. There is no certain safeguard against the gland being overloaded and so producing an unnecessarily tight packing. Consequently, in the case of lightly loaded spindles, the above-mentioned overloading may bind the spindle to the point of stopping its movement.

This possibility occurs in the case of lightly loaded relief valves, or where a slight pressure difference, on a valve, has to produce a spindle movement. Also where a shaft in transmitting a comparatively small torque, the binding of the shaft by a stuffing box might result in an unnecessarily large frictional loss. In such cases it is advisable to consider the possibility of omitting the stuffing box and depending upon a small clearance, between the spindle and cylinder, to keep the leakage within a permissible amount.

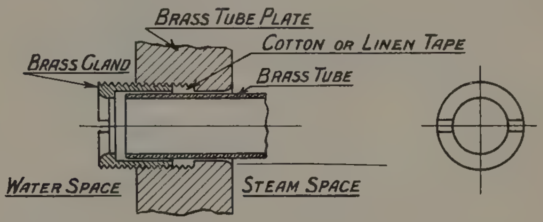

FIG. 5

Estimating Spindle Leakage. Before deciding to omit the stuffing box from a cylinder we should first make sure that the leakage past the spindle, having no stuffing box, will not result in a serious loss.

Suppose the journal bearing for a rotary oil pump has the following dimensions—

Diameter of journal	= 1·5 in.
Length of bearing	= 2·5 in.
Annular clearance between journal and bearing	= ·002 in.
Oil pressure in pump chamber	= 100 lb.-sq. in.

To enable us to estimate the loss of oil past

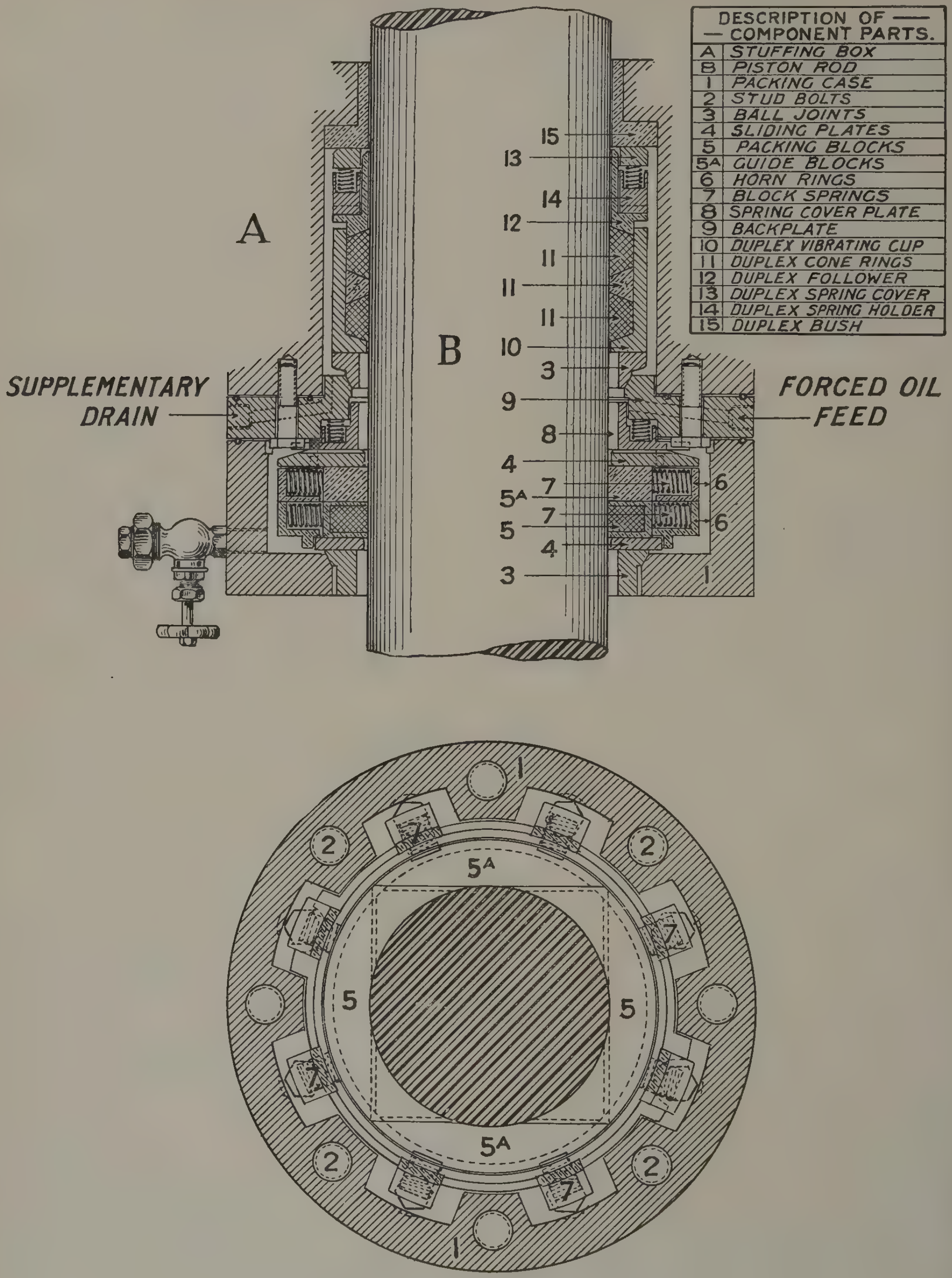

Fig. 6. Stuffing Box with Metallic Packing
(United States Metallic Packing Co., Ltd.)

the clearance, we have the following approximate rule—

$$W = \frac{DS^3P}{12{,}800 L \mu}$$

where

W = leakage in lb. per minute.
D = diameter of spindle in in.
S = annular clearance in thousandths of an inch.
P = pressure difference at spindle ends in lb.-sq. in.
L = depth of spindle covered by cylinder in in.
μ = coefficient of viscosity of fluid in dynes seconds per sq. cm.

Applying this rule to the above example, we have $W = \dfrac{1{\cdot}5 \times 2^3 \times 100}{12{,}800 \times 2{\cdot}5 \times 1{\cdot}9} = 0{\cdot}02$ lb. per minute leakage, assuming the oil used to be American spindle and the working temperature to be 60° F.

This quantity of oil, compared with the pump output, is negligible, and we may confidently omit a stuffing box from the design.

Enlarging or Reducing Drawings. Where a special machine part, such as the metallic stuffing box in Fig. 6, is fitted to a machine, the makers' drawing may be produced to a scale different from the one adopted in the assembly drawing. Further, where the guidance is a photograph or print from a block the scale may

TABLE I

REFER TO FIG. 1

Diameter of rod D = ...	In. 1	In. 2	In. 3	In. 4	In. 5	In. 6	In. 7	In. 8	In. 9	In. 10
Packing thickness T = ...	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
0–100 lb.-sq. in. L = ...	$1\frac{7}{8}$	$2\frac{3}{4}$	$3\frac{11}{16}$	$4\frac{5}{8}$	$5\frac{1}{2}$	$6\frac{3}{8}$	$7\frac{5}{16}$	$8\frac{1}{4}$	$9\frac{1}{8}$	10
100–300 in. L = ...	2	3	4	5	6	7	8	9	10	11
0–300 in. M = ...	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4	$4\frac{1}{2}$	5	$5\frac{1}{2}$

TABLE II

Refer to Fig.	Rod Movement	Application	Remarks
1	Reciprocating of moderate speed	Vertical steam reciprocating engine. Steam cylinder of direct-acting pump	
1	Combined reciprocating and rotary. Slow and irregular	Steam or water regulating valves	
2	Slow and infrequent	Hydraulic machines	Packing load on rod is automatically proportionate to the pressure in cylinder
3	Reciprocating or rotary of moderate speed and low press difference	Lightly loaded or sensitive relief valves. Rotary pumps of small hydraulic head	
4	Very slow and infrequent	Straight steam pipe line	Allows for increase in pipe length due to temperature rise
5	Very small and very slow	Steam surface condenser tubes	Allows for increase in tube length due to temperature rise
6	Reciprocating high speed subject to vibration	Steam reciprocating engines of all speeds	Packing hugs the rod without restraint

be quite odd. In embodying the part on a drawing to a standard scale it is usual to use a pair of proportional compasses, as shown in Fig. 7. In using this instrument, we take a length from the guidance drawing with one pair of legs and, the instrument being adjusted, the corresponding length to the second scale is automatically provided by the span of the opposite pair of legs.

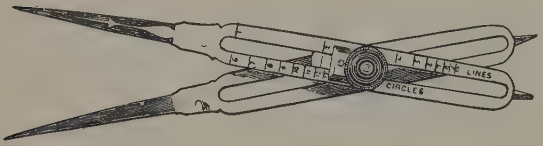

Fig. 7. Proportional Compasses

As an alternative to the use of the proportional compasses, we may make a special scale on a strip of cartridge drawing paper. This can be quickly done as follows.

Referring to Fig. 8, the length *AB* taken from a guidance is to quite an odd scale and, say, shorter than the corresponding length on the scale to be used in the reproduction. We must, therefore, enlarge the view. Suppose we have to reproduce to a scale of $\frac{1}{2}$ in. = 1 in., first marking out, from a standard scale, the divisions of the scale $\frac{1}{2}$ in. = 1 in. along the line *AC*, we then join the points *B* and *C* and draw straight lines through the points marked 5, 4, etc., on *AC* parallel to *CB*.

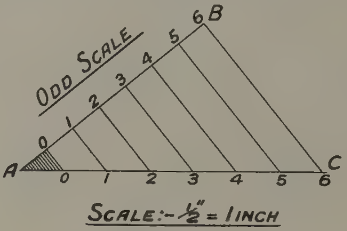

Fig. 8

EXERCISE

Make a scale suitable for enlarging the drawing of the stuffing box, Fig. 6, to a scale of $\frac{1}{2}$ in. = 1 in., and with the aid of the scale proceed to make a drawing of the detail to this scale. Assume the diameter of the piston rod to be 7 in.

GEARING

By Henry E. Merritt, D.Sc.

LESSON VIII

BEVEL GEARING

The tooth action of bevel gears is, in many respects, closely analogous to that of spur gearing, and this tends to facilitate the study of the subject. At the same time, having regard to the great diversity of the possible forms of tooth which can be employed, and the larger number of problems which arise for solution, the subject is rather more complex than that of spur gears, and only the most superficial survey can here be attempted.

The Hypothetical Crown Wheel. It was shown in Lesson I that the pitch surfaces of bevel gears are conical in form, with a common apex at the point of intersection of the shaft axes. In the case of spur gearing the rack member has a flat pitch surface forming part of a plane of unlimited extent; but in the case of bevel gears the corresponding member is the "crown wheel," which has a pitch surface in the form of a flat disc. Fig. 39 shows the conical pitch surfaces of a pair of bevel gears respectively, engaging with opposite sides of the

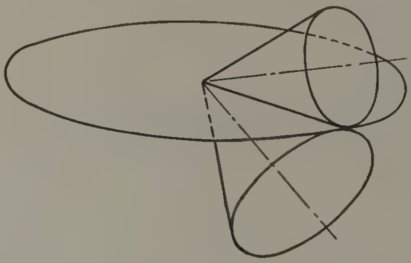

Fig. 39. Rolling of Bevel Pitch Cones on Crown Wheel

pitch surface of the crown wheel, in the same way that a pair of spur or spiral gears may engage opposite sides of the pitch surface of a rack.

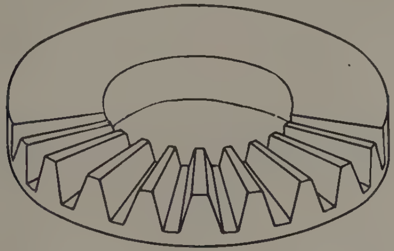

FIG. 40. "OCTOID" CROWN WHEEL WITH PLANE-SIDED TEETH

Extending the analogy, when a pair of spur or spiral gears are in mesh, a hypothetical rack can be imagined to be located between them, the engaging surfaces of the gears and the hypothetical rack all making contact at a common point. Thus, in the rack-shaping process, the cutter sweeps out the surface of the hypothetical rack teeth, and the latter during the rolling motion of the blank generate the correct profile on the gear to be cut. In the same way, if a cutter or cutters be made to sweep out the surface of the teeth of a hypothetical crown wheel, and a bevel gear blank be rolled on its pitch surface with the desired conical rolling motion, teeth will be generated upon it. This forms the basis of all processes of generating bevel gear teeth, although a certain proportion of bevel gears are still produced by a process based on the use of a former of approximately the desired tooth profile.

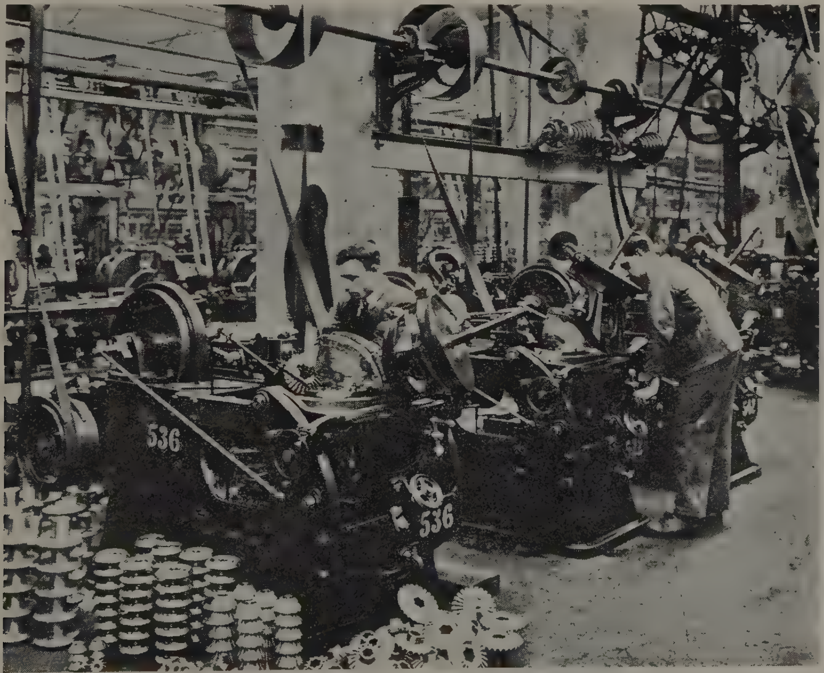

FIG. 41. BEVEL GEAR SHAPING MACHINE
(David Brown & Sons, Huddersfield)

Generating Straight-toothed Bevel Gears. In order to generate a pair of bevel gears which shall give correct tooth action, i.e. transmit uniform velocity, the only condition is that the cutting tools shall describe opposite sides of the surface of the hypothetical crown wheel common to the gears. Accordingly, the system in general use employs a hypothetical crown wheel tooth having plane faces which are swept out by straight-sided cutting tools which converge on the apex; this form of crown wheel is represented in Fig. 40. The resulting tooth-form (due to Hugo Bilgram) is, for reasons which cannot here be detailed, termed an "octoid," but differs only by an imperceptible amount from an involute spur gear tooth. Fig. 41 shows a D.B.S. bevel gear shaping machine operating on the principle above described.

Pitch Cones and Back Cones. Strictly speaking, since straight-toothed bevel gears have teeth which radiate from a point (the apex), the tooth form should be examined on a spherical section, having its centre at the apex. In practice, however, the end faces of the blank are made not spherical, but conical, as shown in Fig. 42, and these end faces are termed the "back cones." The surfaces of these back cones are normal to the surfaces of the pitch cones, and they intersect the latter along the "pitch circles" of the gears. The pitch and tooth proportions are then calculated on the pitch circles thus determined, and are measured on the back cones.

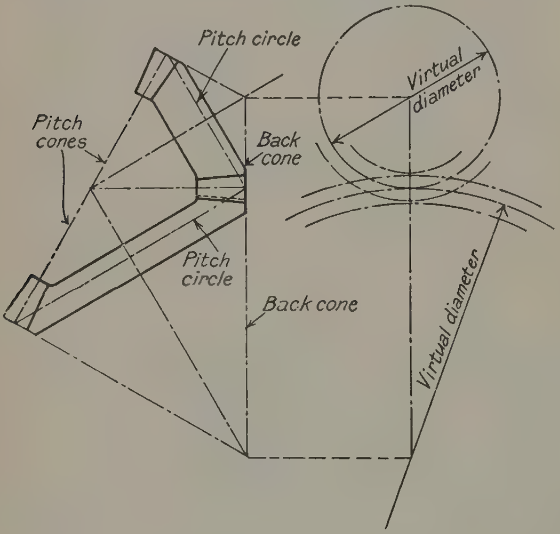

FIG. 42. PITCH AND BACK CONES OF BEVEL GEARS

Virtual Diameters. In considering the shape of the tooth profiles, in connection with problems associated with strength, undercutting, etc., the back cones are "developed," and thus give pitch circles in the form of circular arcs, the radii of which are equal to their respective back cone radii. The diameters of the developed pitch circles are the "virtual" diameters, i.e. the diameters of spur gears having the same pitch and tooth shape. The error involved in making this assumption is negligible. It is thus apparent that a bevel tooth section is always stronger than a spur gear tooth of the same pitch and pitch diameter, since the virtual is always greater than the actual pitch diameter.

Notation and Nomenclature for Bevel Gears. The notation for bevel gear design is an extension of that given in connection with spur gears, and the necessary modifications and additions are as follows (illustrated by Fig. 43)—

Σ = shaft angle.
θ = pitch cone angle.
α = addendum angle = angle between top of tooth and pitch cone.
β = dedendum angle = angle between bottom of tooth and pitch cone.
ϕ = face angle = angle between top of tooth (face of blank) and axis.
ρ = root angle = angle between root of tooth and axis.

These angles are followed by the suffixes $_p$ and $_w$, according to whether they apply to the pinion or the wheel.

C = cone distance or pitch cone radius.
L = wheel tip distance or apex distance = distance between edge of teeth on back cone and the apex.
l = pinion tip distance.

All other symbols as for spur gears.

Detail Design of Bevel Gears. In the detail design of bevel gears it is common to assume the numbers of teeth and the pitch. From these particulars all other dimensions can be calculated.

In order to determine the pitch cone angles from the shaft angle Σ and gear ratio R, the following formula is used—

$$\tan \theta_p = \frac{\sin \Sigma}{R + \cos \Sigma}$$

and when $\Sigma = 90°$

$$\tan \theta_p = \frac{1}{R}$$

and in all cases $\theta_w = \Sigma - \theta_p$

Knowing the pitch cone angle and pitch diameter the back cone can be drawn and the addendum and dedendum, calculated as for a spur gear of the same pitch, can be set off on the back cone. The addendum and dedendum angles can be found from—

$$\tan \alpha_p = \frac{a}{C}$$

$$\tan \beta_p = \frac{b}{C}$$

and the face and root angles from—

$$\phi_p = \theta_p + \alpha_p$$
$$\rho_p = \theta_p - \beta_p$$

The outside diameter of the blank will be—

$$o = d + 2a \cos \theta_p$$

and the apex distance

$$l = C \cos \theta_p - a \sin \theta_p$$

Strength of Bevel Gears. In the case of straight bevel teeth the strength of any cross-section of the tooth is proportional to its distance from the apex. For a given pitch diameter, therefore, the strength does not increase uniformly with increase in face width, and the useful limit to the face width is about one-third of the cone distance. This limit is influenced by the fact that any deflection of the shafts disturbs the load distribution along the teeth, which in any event is difficult to maintain uniform.

It may be shown that within the above limit, assuming load distribution proportional at every point to the strength of the tooth, the ratio of the strength of a bevel gear to a spur of the same pitch and face width is approximately as $C - f$ is to C; thus, if $f = 1/3\ C$, the bevel is two-thirds as strong as a spur gear of the same virtual diameter and face width.

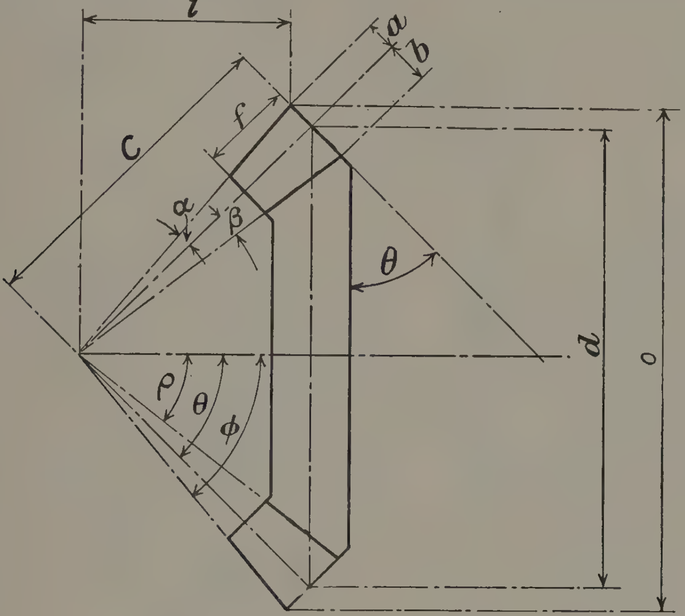

FIG. 43. NOTATION FOR BEVEL GEARS

Spiral Bevel Gears. Recent years have seen a considerable expansion of the field of application of the spiral bevel gear. The chief reason for this is found in the fact that straight-toothed bevel gears are, as a rule, prone to noisy operation when carrying heavy loads at high speeds, whereas the overlapping tooth action of a spiral bevel tends to minimize this defect.

Very many forms of spiral have been and are being employed for the teeth of spiral bevel gears, and it is impossible here to describe any of these in detail. The requirement that the teeth shall be "conjugate" may be satisfied by a large number of curves for the spiral of the tooth, and the spirals which are in practical use are those which lend themselves to easy and rapid manufacture.

(CONCLUSION)

STEAM CONDENSING PLANT

By John Evans, M.Eng.

LESSON VII

STEAM JET AIR EXTRACTORS

The steam jet air ejector is the most reliable and efficient type of air pump, and is used almost exclusively in modern high vacua land condensing plants, having displaced, except for low vacuum work, almost every other type of air pump. Its action depends on the entrainment of air on the surface of a steam jet, and the small space occupied, simplicity, and absence of moving parts form very attractive features.

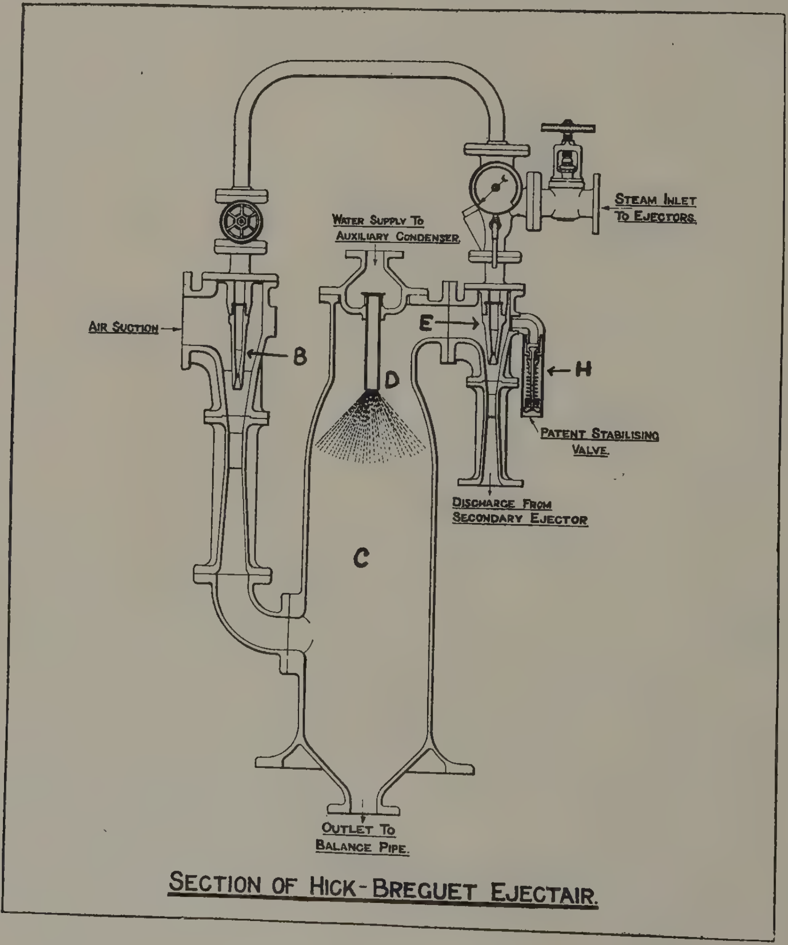
Water Supply To Auxiliary Condenser
Steam Inlet To Ejectors
Air Suction
B
E
D
H
Patent Stabilising Valve
Discharge From Secondary Ejector
C
Outlet To Balance Pipe
Section of Hick-Breguet Ejectair.

Fig. 29

The Hick-Breguet Ejectair, Figs. 29 and 30 (Messrs. Hick, Hargreaves & Co., Ltd.), is the invention of M. Delaporte. It consists of first and second stage single steam jets, and an intermediate or auxiliary condenser. Air drawn from the main condenser is entrained on the surface of a steam jet issuing from the nozzle *B*, the velocity energy of the mixture being converted into pressure energy in the diverging portion of the tube. The mixture is discharged into the auxiliary condenser *C* in which a water spray *D* condenses the steam from the first or primary ejector, and cools the air which is finally withdrawn and discharged to the atmosphere by the second stage or secondary ejector *E*. If the quantity of air passing to the ejectair is reduced owing to light steam loads on the main condenser, the vacuum in the auxiliary condenser (about 4 in. less than the main vacuum), tends to rise, causing the secondary steam jet to *break back*, that is, turbulent and interrupted discharge occurs. Under these conditions the spring-controlled stabilizing valve *H* opens, admitting air to the secondary ejector, and ensuring stability of working under all conditions of load.

The auxiliary condenser vacuum, being less than the main condenser vacuum, permits the use of condensate from the latter as the cooling water supply to the former, the necessary flow being produced by the discharge head of the condensate extraction pump and the auxiliary condenser vacuum.

Fig. 31 shows a sectional arrangement of a typical Hick-Breguet condensing plant. The auxiliary condenser cooling water is drained to the extraction pump branch by means of a "balance pipe" *B*, which also provides a balance of pressure between the main and auxiliary condensers.

When condensate from the main condenser is the cooling water supply to the auxiliary condenser, the ejectair is, in effect, a feed water heater, and, if as in the arrangement shown, the secondary ejector exhaust is led into a feed heater, nearly the whole of the heat in the ejectair operating steam is utilized for feed heating.

The "Compactum" Ejector (Contraflo Co.) is similar in action to that just described. The auxiliary condenser is of the surface type, and the secondary ejector discharges into a second condensing department, usually under atmospheric pressure, both condensers being contained in the same shell. The drainage chambers are incorporated in the same casting as the main condensate inlet and outlet chambers, Fig. 32, and are each provided with a float-controlled discharge valve, through which drainage water is delivered into the main condenser.

The steam jet ejector is, in effect, an air compresser, and the greater the number of stages with intercooling, the less will be the consumption of operating steam. While a two-stage ejector is quite economical for vacua up to 29 in., the steam consumption can be considerably reduced for higher vacua by the adoption of a three-stage unit (Fig. 33).

The Radojet Air Pump. Another form of steam jet air extractor is the Radojet, manufactured by John Musgrave & Co., Ltd. It is of American origin, and derives its name from the form of steam jet and diffuser employed.

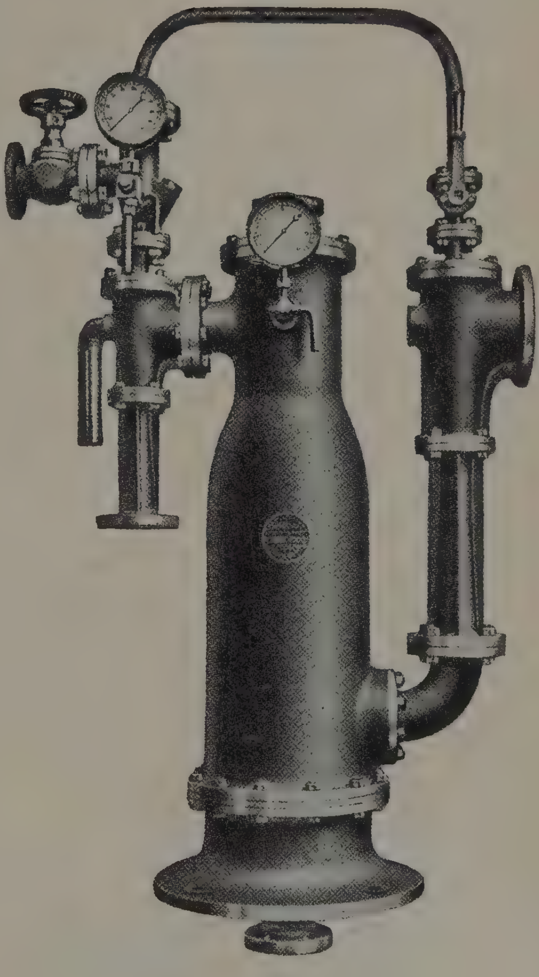

FIG. 30. THE HICK-BREGUET EJECTAIR
(Jet Type)

The first stage, which augments the vacuum produced by the second stage, is of the usual type, having a number of steam nozzles and a tubular diffuser. Referring to Fig. 34, live steam enters the Radojet through a stop valve

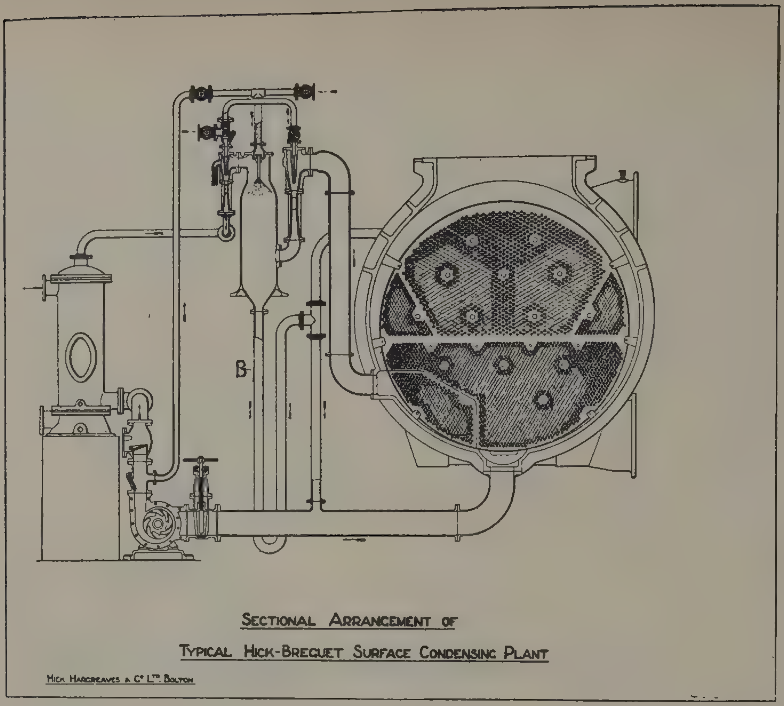
B
SECTIONAL ARRANGEMENT OF
TYPICAL HICK-BREGUET SURFACE CONDENSING PLANT
HICK HARGREAVES & Cº Lᵀᴰ BOLTON

FIG. 31

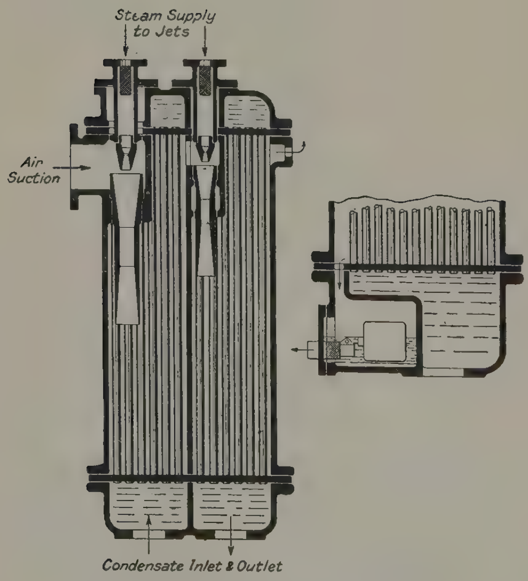

FIG. 32. "COMPACTUM" AIR EJECTOR
(Two-stage)

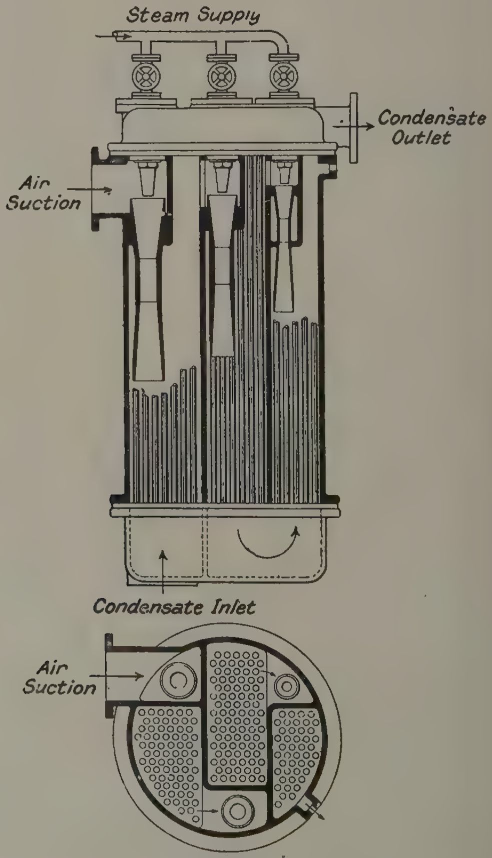

FIG. 33. "COMPACTUM" AIR EJECTOR
(Three-stage)

and strainer 1, which is connected to the first and second stage nozzles 2 and 3 respectively. Steam expands through the first stage nozzles 2, and entrains air and vapour entering through the suction branch 4. The mixture of air and steam passes down the diffuser 5, and is discharged to the suction chambers 7 of the second stage, the absolute pressure of the mixture having been raised by several inches of mercury by compression in the diffuser 5. The remainder of the steam entering at 1 passes through the second stage nozzle 3, which has a constant throat area, but a variable ratio of expansion. This variation is accomplished by the nozzle point 8, and can be made while the Radojet is in operation, by means of the extension of the point which projects from the casing at 10.

The steam issuing at 9 from the nozzle 3 in the form of an annular sheet, having a very large area compared with the weight of steam passing, entrains the steam and air mixture from the suction chamber 7, and compresses it in the annular diffuser 11 to atmospheric pressure. The mixture then passes to the discharge belt 12, and is exhausted to atmosphere through the opening 13.

By arranging a small surface condenser between the two stages (Fig. 35), the second stage has only to deal with air and incondensable gases, the steam from the first stage being condensed in the auxiliary condenser and removed by a vacuum trap or condensate lifter. For this reason the total live steam consumption of any two-stage steam jet ejector is approximately halved when an intercondenser is fitted. It will be observed that in the Radojet, the cooling water for the auxiliary condenser surrounds the tubes, and the air and vapour mixture flows through them. Although this is contrary to usual condenser practice, it is a distinct advantage, as it ensures the high air velocity essential in order to obtain a reasonable heat transmission rate through the auxiliary condenser tubes.

The Weir Air Ejector. Fig. 36 shows the two-stage intercondenser type of air ejector as made by Messrs. G. & J. Weir, Ltd. Its action is similar to that of the two-stage ejector previously described, the arrangement of the ejectors and auxiliary condenser being slightly modified. The adoption of the surface type of auxiliary condenser makes it specially suited for marine purposes, where a circulation of sea water is arranged. This is essential, because during manoeuvring, when the main engines are liable to be suddenly shut down, the air ejector must maintain the vacuum in the main condenser. If condensate is circulated through the auxiliary condenser a supplementary cooling water supply must be provided.

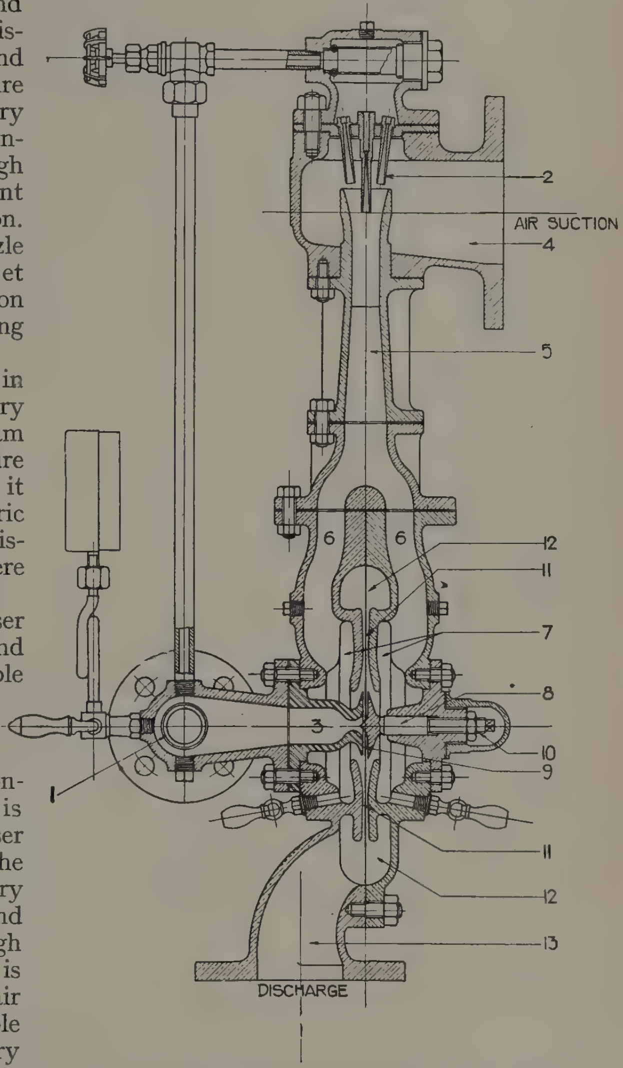

Fig. 34. Musgrave "Radojet" Air Pump

Kinetic Rotary Air Pump. High velocity water jets may be employed for producing a vacuum, but the highest vacuum obtainable is determined by the temperature of the water. Also, as the air removing capacity of any form of jet extractor depends on the kinetic energy of the jet, it follows that for high vacua and

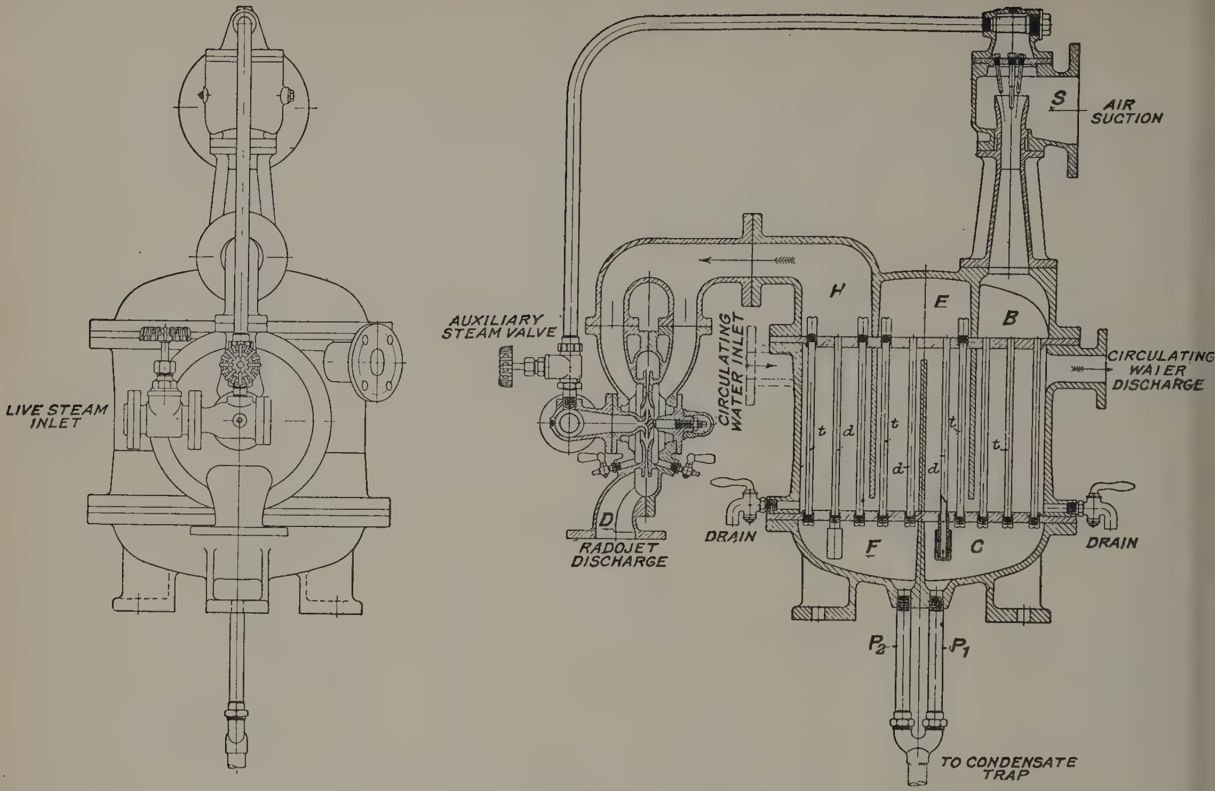

Fig. 35. Section of Intercondenser Radojet

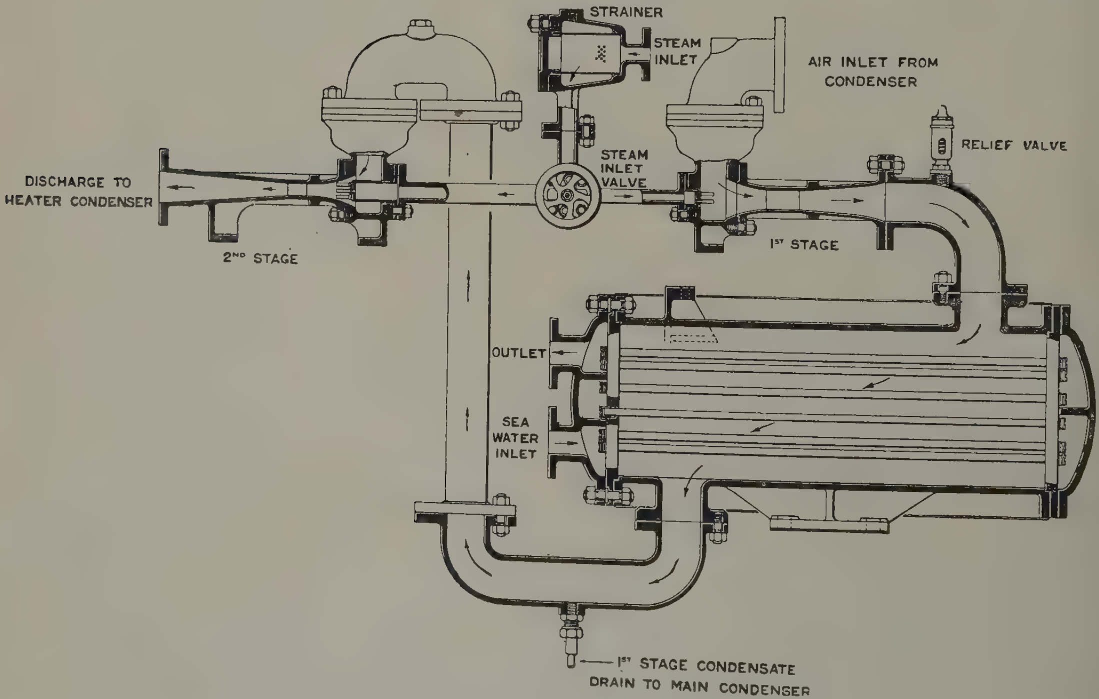

Fig. 36. Weir Air Ejector

large air duties the quantity of water and its pressure become very large, and the cost of power excessive. This can be overcome, and the efficiency increased, if a steam jet first removes the air from the condenser and delivers it to the water jet after having increased its pressure.

The three chief parts of this type of air extractor are—

1. A steam jet and diffuser to partially compress the air.
2. A water jet and diffuser to complete the compression.
3. A pump, usually of the centrifugal type, to impart sufficient pressure to the water to produce a high-velocity water jet.

The Contraflo Co. kinetic air pump (rotary system), Fig. 37, operates on this principle, the steam jet ejector forming the first stage, and the water jet the second. A steam ejector *A* withdraws the air from the condenser, discharging it into an auxiliary condenser *C*, where the operating steam is condensed by condensate circulated by the extraction pump *D*, and discharged to the boiler feed pumps. The cooled air and its associated vapour pass through a non-return valve *F* to a water jet operated ejector *K*, the operating water being supplied by a rotary pump *H* to an annular water nozzle *L* and a central solid nozzle *K*. The combined water and air from the nozzles *K*

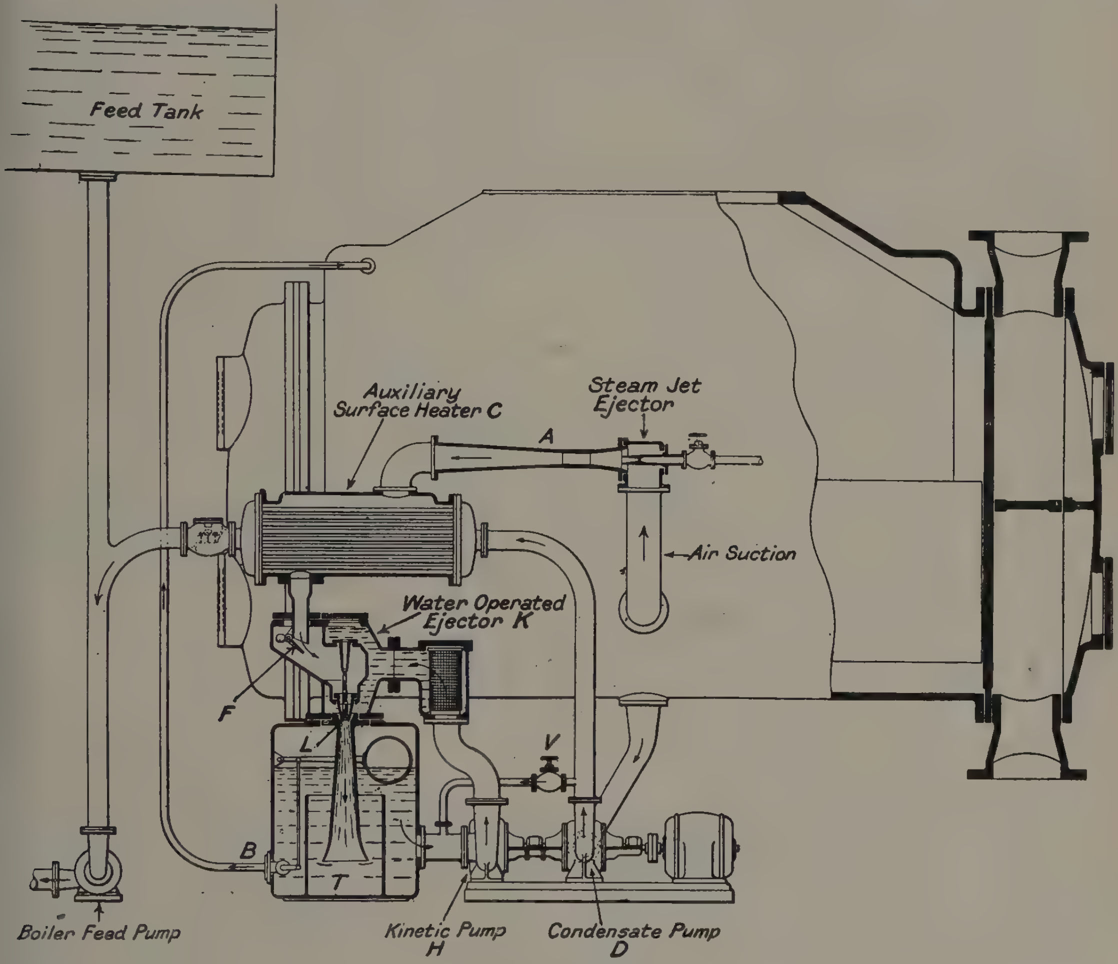

Fig. 37. "CONTRAFLO" KINETIC AIR PUMP
(Rotary System)

(*Continued on page* 712)

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.I.N.A.

LESSON IV

CLASSIFICATION OF STEAM TURBINE TYPES—(*contd.*)

Pure Reaction Turbine. This type, which is illustrated diagrammatically in Fig. 13, comprises two or more radial tubes rotating on and in communication with another pipe through which steam is supplied to nozzles which are attached to the ends of the radial pipes, and arranged with their axes in the tangential direction.

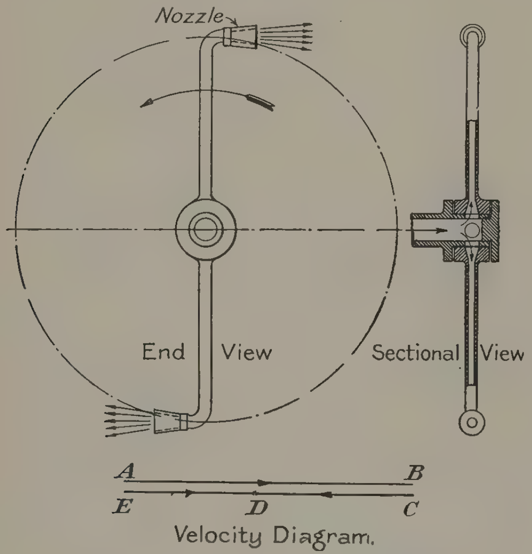

FIG. 13. PURE REACTION TURBINE

Steam expands in the nozzle and issues with a certain velocity which is represented by the vector AB. If CD represents the peripheral velocity of the nozzle, then ED will represent the velocity of the steam relative to a fixed point at a given instant. Due to the change of velocity which occurs, there is a thrust or *reaction* on the rotating tube and tangential to it. This reaction constitutes the driving force, hence the name. In view of the fact that this type of turbine has no large representatives, we shall devote no further space to it.

Impulse-reaction Turbine. This important type was developed by the Hon. Sir Charles A. Parsons. The essential difference between the type under consideration and the pressure compounded impulse type is that, whereas the steam pressures at the inlet and outlet sides of the impulse blade are practically equal, *the steam definitely undergoes expansion in the moving blades of the impulse-reaction turbine as well as in the fixed blades.* In all other respects, although constructional details differ in the two types, the principles of operation are the same.

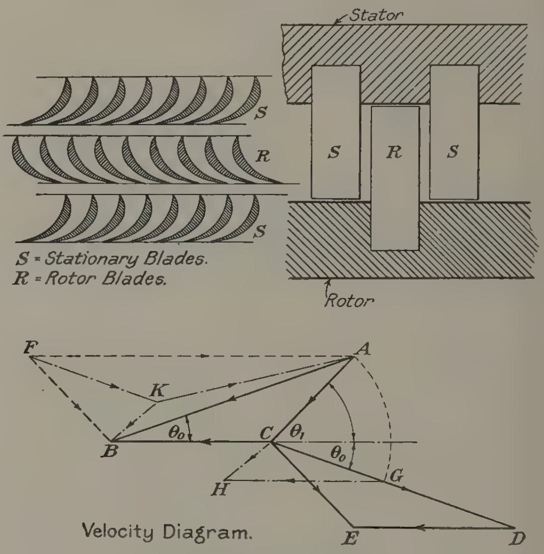

FIG. 14. DIAGRAM OF IMPULSE-REACTION (PARSONS) TURBINE

The total expansion of the steam from initial to exhaust pressure is carried out in a large number of stages arranged in series. Each stage may be imagined to consist of a complete ring of fixed guide blades (nozzles) which are attached to the turbine casing, and a complete ring of moving blades (nozzles) which are attached to the rotor, which for the present may be imagined to be a rotating steel drum. (See Figs. 14 and 15.) The blades are usually identical in shape, and the clearances between the fixed blades and the rotor and between the moving blades and the casing are made as small as practicable.

The amount of expansion of the steam in the various blades will depend mainly on the net area of the whole of the blade channels in any one ring, and it follows, therefore, that if the blades are made identical in shape, then, apart from small changes in the specific volume of the steam, the amount of expansion which the steam undergoes in two adjacent blade rings (one fixed, the other moving) of the same radial height will be the same. When these conditions are fulfilled, the velocity of the steam *relative to the blades at outlet* will be the same in the moving blades as in the fixed blades, and this condition is referred to as *half-degree reaction.*

Consider the velocity diagram shown in Fig. 14. Let AB represent the velocity of the steam issuing from a group of stationary blades, and let CB be the mean peripheral velocity of the rotor blades. Then AC represents the velocity of the steam relative to the moving blades at inlet. Suppose now that there is no change of pressure in the rotor blades and that there is no loss of kinetic energy, due to friction or other cause. Then CG would represent the exit velocity relative to the blade, and this compounded with the velocity GH, which is the blade velocity, would give the velocity CH of the steam relative to the casing, at outlet. If KB be drawn equal and parallel to GH, KA will represent the change of velocity suffered by the steam in a pure impulse turbine equipped with blades of this section. The thrust on the blades due to *impulse* will be proportional to and in the same direction as AK.

We have stated above that in the impulse-reaction turbine with half-degree reaction, the steam expands to such an extent that the exit velocity CD is, to all intents and purposes, equal to the initial absolute velocity AB. The increase of velocity from CG to CD and the gain of kinetic energy equal to

$$\frac{CD^2 - CG^2}{2g} \text{ ft.-lb. per lb. of steam}$$

is due to *reaction.*

By combining the velocity CD with the blade velocity DE, we obtain the final absolute velocity CE, which is, of course, the velocity with which the steam flows into the next row of fixed blades. Drawing FB equal and parallel to CE, and joining A and F, we obtain the change of velocity FA, suffered by the steam in the actual turbine. The component FK represents the additional change of velocity (and to a certain scale, the additional thrust) due to *reaction,* FK being equal and parallel to GD.

Efficiency of Impulse-reaction Turbine.

Let V_1 = absolute velocity of steam at exit from the fixed blades (Fig. 16).

,, u = blade velocity.

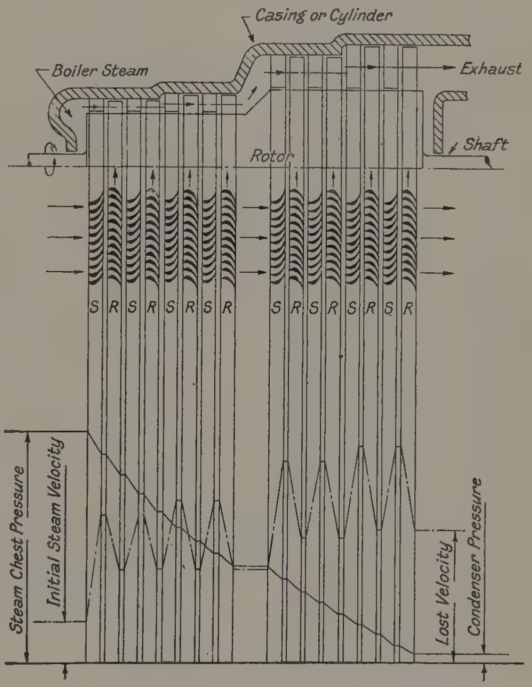

FIG. 15. DIAGRAMMATIC ARRANGEMENT OF AXIAL FLOW IMPULSE-REACTION OR "REACTION" TURBINE

S — Stationary blades.
R — Rotor blades.

Then V_{ri} (AC) represents the velocity of the steam relative to the moving blades at inlet. Similarly, V_r (CE) represents the absolute velocity of the steam at exit from the moving blades, and hence the relative velocity of the steam at inlet to each ring of fixed blades. Thus, to each ring of blades (fixed and moving alike) the inlet velocity of the steam is V_{ri} ($=V_r$) and the outlet velocity is V_1. Hence, the gain in kinetic energy, which occurs in each blade ring due to the expansion of the steam, must be

$$\frac{V_1^2 - V_r^2}{2g} \text{ ft.-lb. per lb. of steam,}$$

and the total energy supplied, assuming no losses, in a stage comprising one fixed and one moving ring of blades equals

$$\frac{V_1^2 - V_r^2}{g} \text{ ft.-lb. per lb. of steam.}$$

Since the triangles ABC, CDE are congruent, the angles θ_1 and β are equal, and further, since $FB = CE$, AF is parallel to BC.

Hence, AF is the velocity of whirl and is equal to

$$\begin{aligned} V_w &= V_1 . \cos\alpha + V_r . \cos\beta \\ &= V_1 . \cos\alpha + V_{ri} . \cos\theta_1 \\ &= V_1 . \cos\alpha + V_1 . \cos\alpha - u \\ &= 2V_1 . \cos\alpha - u. \end{aligned}$$

∴ Thrust on blades per lb. of steam per second

$$= \frac{2V_1 . \cos\alpha - u}{g} \text{ lb. weight.}$$

$$\text{Work done per lb. of steam} = \frac{u(2V_1 . \cos\alpha - u)}{g}$$

and, writing $\rho = \dfrac{u}{V_1}$, as before,

$$\text{Work done per lb. of steam} = \frac{V_1^2}{g}(2\rho . \cos\alpha - \rho^2)$$

$$\begin{aligned} \therefore \text{ efficiency} &= \frac{\text{work done}}{\text{energy supplied}} \\ &= \frac{V_1^2}{g}(2\rho . \cos\alpha - \rho^2) . \frac{g}{V_1^2 - V_r^2} \\ \text{i.e.} &= \frac{V_1^2(2\rho . \cos\alpha - \rho^2)}{V_1^2 - V_r^2} \end{aligned}$$

It will be shown later that the efficiency is a maximum when $u = V_1 \cos\alpha$, and if we assume that α is 20°, as it actually is in the majority of Parsons turbines, then $V_r = V_1 \sin \alpha = 0{\cdot}342\, V_1$. It follows that V_r^2 is about 12 per cent of V_1^2.

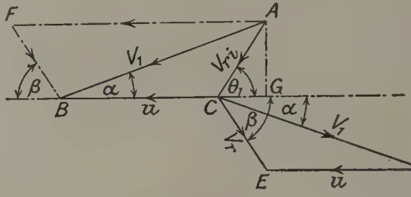

FIG. 16

If we neglect V_r^2, the efficiency is given by the simple expression

$$\eta = 2\rho . \cos\alpha - \rho^2.$$

$$\text{and} \quad \frac{d\eta}{d\rho} = 2 \cos\alpha - 2\rho.$$

For a maximum value of the efficiency, $\dfrac{d\eta}{d\rho} = 0$, and therefore

$$\rho = \cos\alpha.$$

If we assume $\alpha = 20°$, $\cos\alpha = 0{\cdot}94$, and we see that for maximum blade efficiency, the blade speed is very nearly equal to the steam speed.

NOTE. For those readers still unacquainted with the calculus the above result, obtained by differentiation, may be obtained by plotting values of η against values of ρ. In this case, however, it is difficult to judge just what value of ρ corresponds to the maximum value of η.

STEAM CONDENSING PLANT

(*Continued from page* 709)

and L are discharged through a diffuser, where the kinetic energy of the jet is converted into pressure energy, thus allowing the water to be discharged against atmospheric pressure in the tank T.

In order to prevent a gradual rise in temperature in the tank by the drainage water from the auxiliary condenser, a small quantity of water is allowed to flow from the condensate pump discharge through the valve V to the suction of the rotary pump. The surplus water in the tank T is automatically by-passed into the main condenser by means of the float controlled valve B. A circulation around the tank and main condenser is set up, the temperature of the water supply to the ejector being kept sufficiently low to ensure efficient operation and unimpaired vacuum.

PUMPS: CONSTRUCTION AND MAINTENANCE

By Owen A. Price, M.I.Mech.E.

LESSON VII

CENTRIFUGAL PUMPS

eneral. During the discussion on " Head " in esson II, it was pointed out that the velocity ead (*h* ft.) possessed by a stream of water owing at a velocity *v* ft. per second is, numeri- ally, equal to the vertical height which a body vould attain if projected vertically upwards vith velocity *v*.

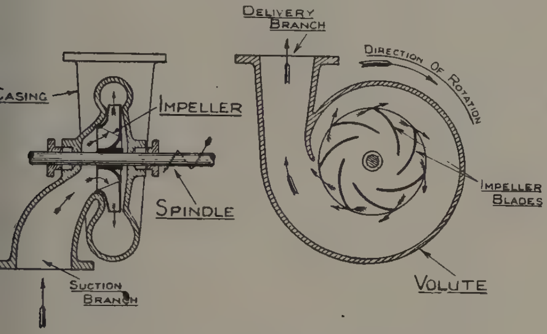

Fig. 25. Elementary Centrifugal Pump

The seventeenth century notion of the working f the centrifugal pump was (almost in Denis Papin's words): If the water were projected angentially from the tip of the rotating blade t a certain velocity, it would be carried to the eight which bodies can reach when thrown ertically upwards at the particular velocity in uestion ; also, if the tangential velocity were reater or less, the height reached would be reater or less in the ratio of the squares of the elocities.

That is to say, the conditions are similar to hose explained for the water column of Fig. 7A, r, the kinetic energy at the bottom of the olumn is equal to the potential energy at the op of the column. Thus, the height *h* of the vater column supported by a centrifugal pump mpeller is equal to $\frac{v^2}{2g}$, where v is the velocity f the water at the impeller periphery.

This time-honoured and exceedingly useful vay of considering the head produced by a entrifugal pump gives a definite physical coneption of the behaviour, and has led to most of the practical designing rules employed by manufacturers of these machines.

Simple Centrifugal Pump. In its elementary form a centrifugal pump (Fig. 25) consists of three parts—

1. A revolving vane or *impeller.*
2. A casing consisting of a collecting chamber or *volute,* with the necessary inlet (suction) and outlet (delivery) branches, and
3. A driving *spindle.*

Action. The pump being filled with water and the impeller rotated, water is thrown out centrifugally from the periphery of the impeller into the volute. This action tends to produce a vacuum at the centre, or *eye,* of the impeller, with the result that fresh water is forced up the suction pipe (by atmospheric pressure) to take the place of the ejected water. Provided the speed of rotation is suitable to the working conditions, and assuming that the water column is not ruptured by air leakage or by vapour formation, due to excessive suction lift, a steady rate of delivery is maintained so long as the impeller is kept in rotation.

There are no valves in a centrifugal pump, therefore the condition of pressure and velocity supported by the impeller is due entirely to the peripheral velocity of the impeller. We shall show later that a rotating impeller creates a *pressure-difference* (or hydraulic tension) between the bodies of water at its eye and its periphery, quite regardless of what the absolute pressure may be at either of these two points. Therefore, if water is passed through *n*, similar impellers arranged in series, or in *stages,* it will emerge at a pressure equal to *n* times the pressure created by a single impeller.

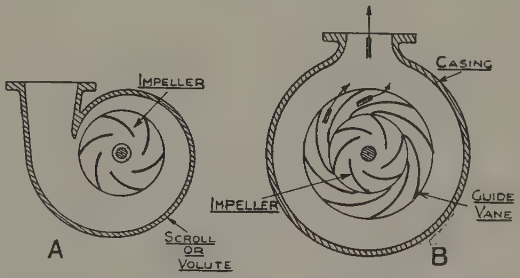

Fig. 26. Volute and Guide Vane Pumps

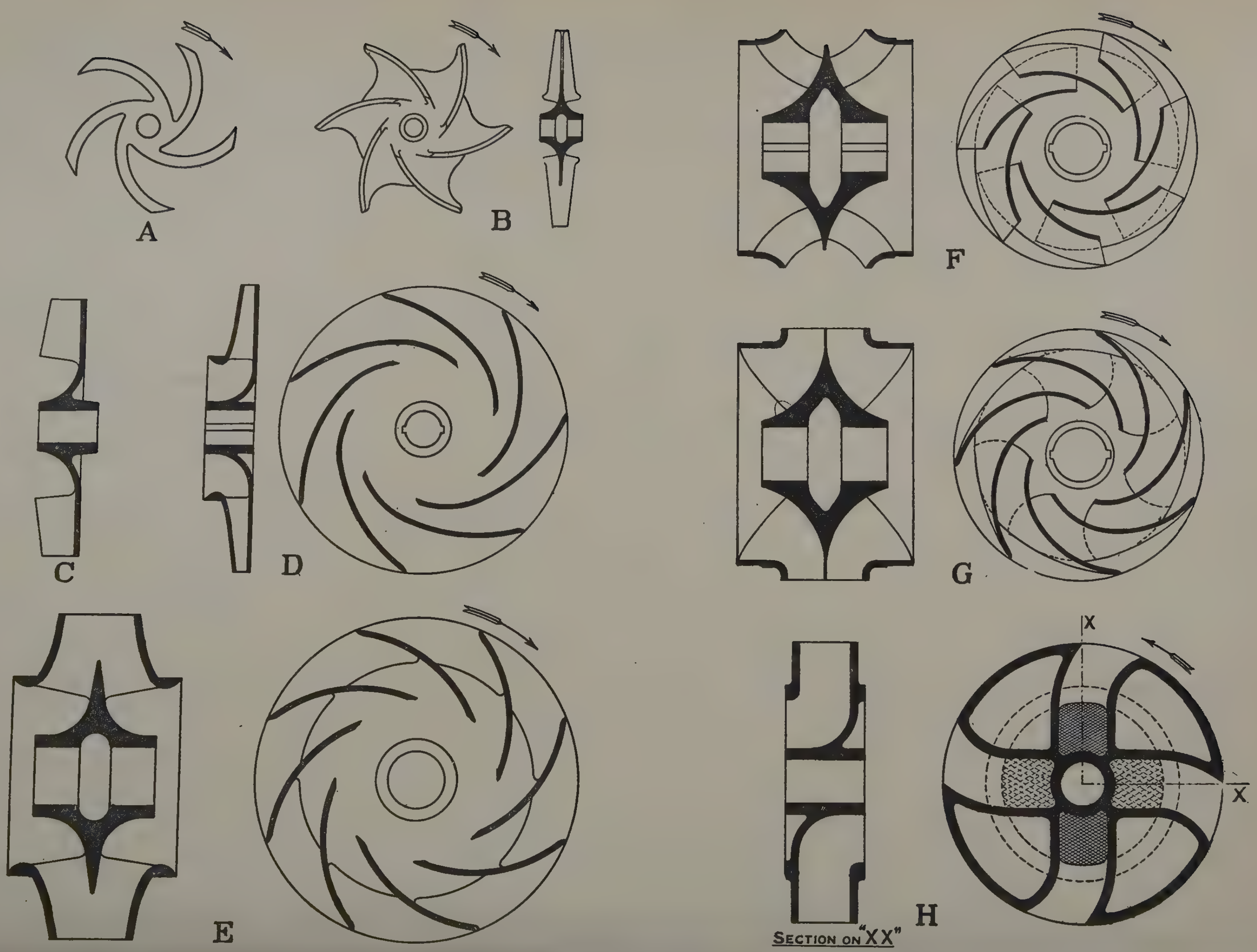

FIG. 27. PUMP IMPELLERS

Classification. Centrifugal pumps may be grouped into two main divisions: *volute pumps*, Fig. 26 (*A*), in which a simple collecting chamber. scroll, or spiral housing surrounds the impeller, and *guide vane pumps*, Fig. 26 (*B*), in which a number of specially formed diffusion passages are disposed around the impeller.

Either form may be built up as a *single-stage* pump, may be connected in series as a *multi-stage* pump, or arranged in parallel as a *parallel* pump.

From the constructional point of view we may recognize many varieties, such as *horizontal-spindle* and *vertical-spindle* pumps, *single-suction* and *double-suction* pumps, *end-cover* and *split-casing* pumps, and so on.

It is apparent, therefore, that the distinctions between centrifugal pumps are largely matters of detail construction, and we shall now consider the common forms adopted for the essential parts.

VOLUTE PUMPS

Impellers. Most of the ordinary varieties of impellers are shown in Fig. 27. *Open type* impellers necessarily run practically in contact with the machined side of the pumping casing. This fact, coupled with strength considerations, limits their application to low speeds and, therefore, to low heads. They have the great advantage of simplicity in manufacture and, to a considerable extent, are self-clearing if solid material passes into the pump.

The *solid arm* impeller, Fig. 27 (*A*), is made in small sizes, where "ruggedness" and durability are more important than high efficiency. The *ribbed form B* is the commoner type of open impeller and, when properly designed, gives very good efficiency, especially on medium and large sized pumps operating against low and moderate heads.

The *open-sided* or half-shrouded impeller *C* is confined chiefly to small units, because it is subject to end thrusts axially. It is also a very useful form for dealing with water containing stringy material, as it possesses the same self-clearing properties referred to for the open type impeller.

The *shrouded* or enclosed form of impeller is exceedingly popular for all classes of centrifugal pump. It is a very strong construction capable of running at high speed, it is easily machined, readily balanced, and can run with considerable clearance from the casing wall. Slip across the sides of the impeller vane is impossible, and analysis of the velocities through a pump will show that the water passing through is subjected to less wall friction than is the case with an open type impeller. The single-inlet shrouded impeller, shown in Fig. 27 (*D*), is commonly used for ordinary small centrifugal pumps, and universally used for multi-stage turbine pumps for all heads. Example *E* shows the customary construction for double-inlet impellers, either for high or low head and for both large and small pumps.

To meet the case of quick rotary speed on low head the impeller diameter must be made as small as possible. Examples *F* and *G* show two methods of attaining this object, *F* is a very shallow bladed impeller of special form, and *G* is a twisted vane impeller with "Francis" blades, similar to those employed on mixed-flow water turbines. Both forms may be carried out either as single-inlet or double-inlet impellers.

Centrifugal pumps are frequently used for pumping liquid containing mud and stones, small coal, fallen leaves, grass, and water weeds, the refuse to be found in crude sewage, etc., etc. When the pump is of sufficiently large dimensions it will perform these duties without modification in design. Smaller impellers, however, are very liable to choke, and many special designs have been advanced for avoiding this contingency. For many years one of two general principles has been followed, viz., to provide a cutting knife at or near the impeller inlet, or, to employ more or less tubular impeller passages so proportioned that any object which can enter the passage will readily pass through the entire pump. Fig. 27 (*H*) is a double-sided tubular impeller with four passages, two of which open at each side of the impeller. This example gives a general idea of all tubular impellers, though obviously they may be made single- or double-sided, and may be varied considerably in construction.

Casings. The casings of single-stage volute centrifugal pumps may be distinguished by the method adopted for inserting the impeller, and for each method there are usually two possible forms of casing, viz., that for a single inlet and that for a double inlet impeller. Other considerations influencing the style of casing are that certain forms are suitable only for certain types of impeller, or, a particular design may be exceedingly easy to manufacture but may not be of very rigid construction, thus confining its application to low head duties. Another

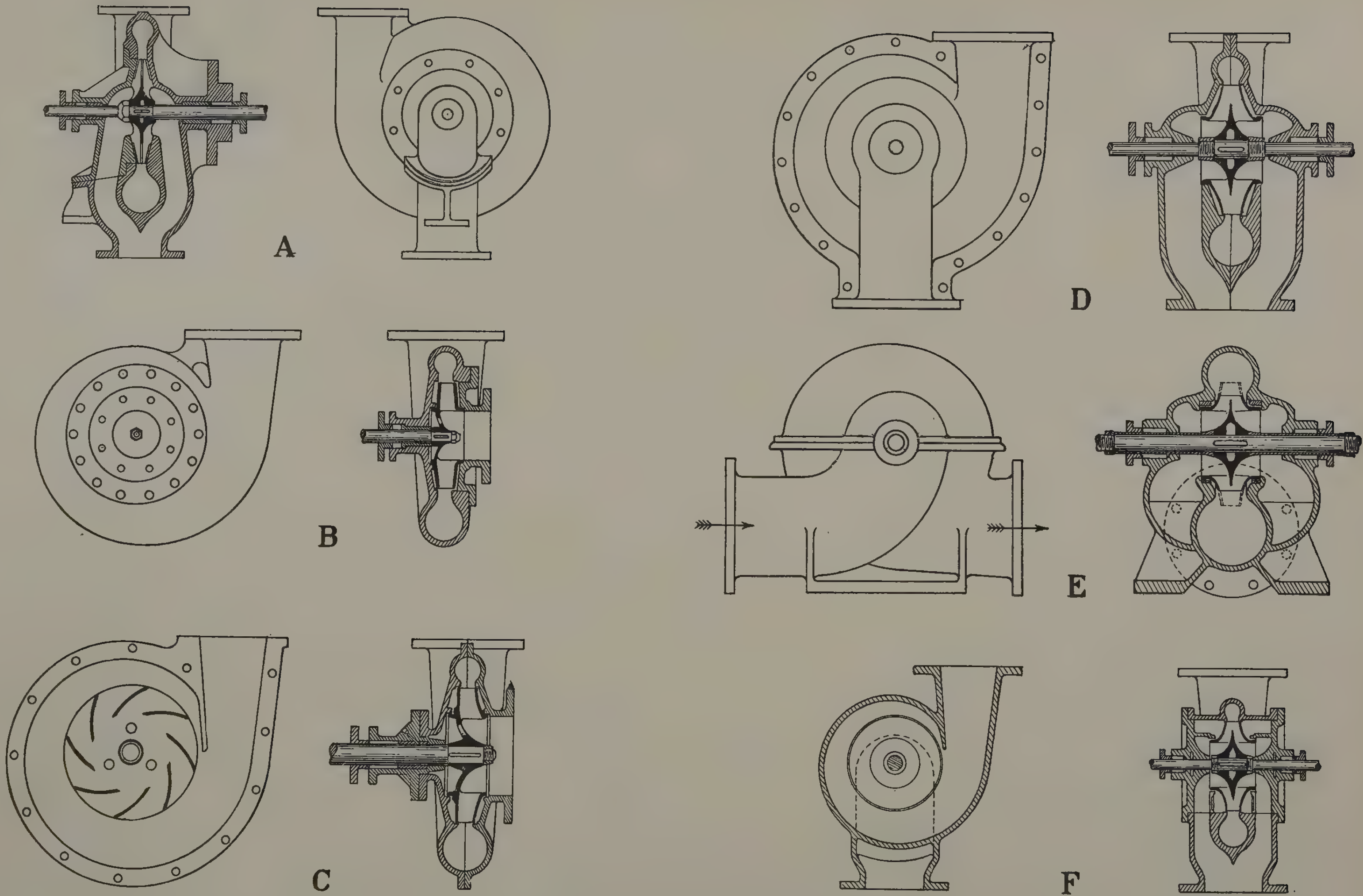

Fig. 28. Pump Casings

type may be a strong design, but rather more expensive in moulding and machining; for high head duties this design would be imperative, and its economy in material might permit of its use even for comparatively low heads.

For the purpose of inserting the impeller all casings must be either provided with an end cover, or the casing itself must be split. Fig. 28 (*A*) shows a single-end-cover form of casing suitable for a double-suction open-type impeller, such as Fig. 27 (*B*). To overcome the difficulty of making a right-angled joint across the suction branch, this cover is coned so that as it is drawn into position it joints in two directions. The jointing material used is fairly thick and composed of red lead putty and twine. The cover and casing surfaces are necessarily machined over a diameter equal to the size of the impeller, but it is usual to *smooth-machine* a narrow contact ring only, corresponding with the inner end of the impeller vanes, and to *rough-machine* the outer portion so that there is a definite clearance between the casing and impeller vanes. This form of casing was originally known as Gwynne's "Invincible" type; it is very suitable for low working heads, and provides ready accessibility for dismantling under working conditions.

With the single-inlet impeller the suction branch is arranged centrally on one side of the casing so that the side-cover joint, Fig. 28 (*B*), becomes a plain flat circular face. The illustration includes a shrouded impeller of Fig. 27 (*D*) type, but it is apparent that with very slight modifications the casing would be suitable for types *A* or *C* on Fig. 27.

The advantages of splitting a casing through the centre of the volute are that a volute core is avoided, and, in the finished casting, the whole of the volute interior is accessible for smooth-dressing. Thus, moulding is cheapened and smooth walls are secured for the passage conveying the highest velocity water in the pump. The machining of such a casing is more costly, because a much larger face must be traversed by the cutting tool, but the handling of the pump is facilitated, because the heaviest casting is made in two parts of approximately equal weight. Examples of this kind of casing are shown in Fig. 28, *C* and *D* respectively, for single-inlet and double-inlet impellers.

Though the type of casing just considered offers distinct manufacturing advantages, it does not provide conveniences for quick dismantling and examination under working conditions. A horizontally divided casing, Fig. 28 (*E*), with both suction and delivery branches arranged in the bottom half casing, probably gives the maximum facility for thorough internal inspection with the minimum amount of dismantling. The production cost of such a type may be greater than for the types previously discussed, but the advantages are with the user rather than the manufacturer.

All the casings so far described are specially suitable for low and medium working heads. A particularly strong form of body suitable for the highest heads is shown in Fig. 28 (*F*). The deep central cylindrical portion of this casing renders it exceedingly strong, and it is a simple type to manufacture.

Feet have been omitted intentionally on most of the pump casings illustrated because, generally, the casing may be used either for a horizontal spindle or a vertical spindle pump, and the feet would be designed accordingly.

FANS AND BLOWERS

THEIR CONSTRUCTION AND OPERATION

By F. G. Whipp, A.M.I.Mech.E., A.M.I.H.V.E.

LESSON V

OPERATION OF PROPELLER FANS

Working Speeds. The speed at which a volume fan should run is first governed by the duty required of it, viz., the class of installation to which it is to be adapted and the quantity of air to be handled.

If the fan is required to work in a factory or workshop building, where silent running is not essential, a fairly high rate of speed may be maintained. A good average speed found in practice to give the best efficiencies is one equivalent to a peripheral velocity of 6,000 ft. per minute. At this rate of travel, however, a hum would be set up which would be disconcerting, in fact, intolerable in a public building, office, or such-like establishment. For silent operation the tip speed of the fan wheel should not exceed 4,000 ft. per minute, but from 3,000 to 3,500 ft. per minute is a safer limit.

Standard Sizes. Propeller fans are constructed in sizes from 9 in. diameter of wheel to 5 ft. diameter in the case of standard makes, but larger fans of almost any practical size can be constructed, reaching to 8 and 9 ft. wheel diameter.

Capacities. Below will be found a useful table giving complete data of the standard range of sizes which has been computed from the information included in the foregoing lessons. The treatment has been generous, and the minimum and maximum outputs have been based upon volumetric efficiencies falling on the upward and downward sweep of the curve, so that slightly better results may be expected for intermediate volumes.

On comparison with the leading makers' capacity tables they compare favourably, and can be employed in practice with confidence.

Mechanical Construction and Proportion. Fig. 11 shows the lines, with proportions, of a typical standard propeller fan arranged for belt driving. Obviously, two bearings are essential, and these are usually supported by a system of bracket arms numbering two, three, or four. When two arms are employed they have to be heavy in order to ensure rigidity, and this method is now almost entirely superseded by the tripod system, viz., three arms set at 120°. These arms are narrow, and so do not obstruct the free area of the fan more than necessary. They are bolted to a foundation ring, usually of tee section, on to which suitable lugs are cast for bolting down the set complete. The bearings are plain grease-lubricated for normal conditions, but where difficulty of access exists ring-oiled or ball bearings are fitted.

When the larger sizes are reached, ring-oiled bearings are often used, but as the speed

Dia. In.	Effective Area	Revs. per Min.		Output Cub. ft. per Min.		Air Horse-power		B.H.P.	
		C = 3,000	C = 6,500	C = 3,000	C = 6,500	Low Speed	High Speed	Mech. Effy. = 33%	Mech. Effy. = 43 %
9	0·414	1,275	2,760	405	860	·004	·0379	·0121	·088
12	0·736	960	2,070	720	1,525	·0041	·0673	·0125	·1565
15	1·150	775	1,650	1,120	2,380	·011	·1052	·033	·245
18	1·656	640	1,380	1,600	3,430	·0159	·1516	·048	·353
24	2·943	480	1,035	2,870	6,150	·0283	·2716	·086	·632
30	4·596	380	830	4,400	9,550	·0432	·4213	·131	·980
36	6·627	320	690	6,350	13,700	·0625	·6068	·189	1·410
42	9·020	270	590	8,800	18,700	·0867	·7987	·265	1·857
48	11·781	240	520	11,450	24,300	·1127	1·0368	·341	2·41
54	14·910	215	470	14,500	30,800	·1426	1·3158	·422	3·06
60	18·407	190	420	17,950	38,200	·1755	1·6307	·532	3·79

in this case is low, such practice is not necessary, unless dictated by other circumstances.

The thrust is taken up by a system of convex discs of hardened steel, or, if preferred, a ball thrust washer may be adapted. The former method is inexpensive and gives excellent results.

Electric Drives. In the case of electrically-driven fans the motor is carried by arms as before, these being specially designed for bolting to the motor casing. Up to about 30 in. diameter of fan wheel no outer bearing is needed, as the wheel is supported by the spindle which already runs in the double bearings of the motor.

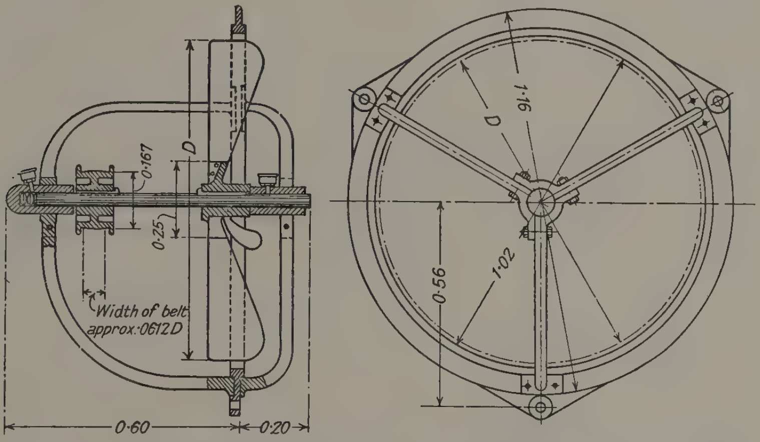

FIG. 11. APPROXIMATE PROPORTIONS OF STANDARD PROPELLER FANS
$D = 1$

The load of a fan is very little until it is revolving at a fair speed, so that it is always capable of maintaining an even balance of power, as the greater the speed the more resistance is offered by the air, and series winding is adopted without any shunt circuit. Up to an absorption of 200 watts a simple double-pole switch is all that is necessary for starting, but above this power a starter should be provided. If variable speed is desired, rheostats must, of course, be included in the circuit. For alternating current work the motors are usually of the short-circuited induction type. In either case the diameter of the motor must be as small as possible, so as not to block up the fan feed and impair its capacity.

Although the driving agent is shown on the feed side of the fan, this can be fitted on the discharge side if circumstances demand, and makers usually list various combinations to suit all conditions likely to be met with in normal work.

Uses. It would be impossible to give anything like a complete list of the uses to which this type of fan can be adapted, as besides being numerous, fresh ones are always being added. To quote a few of the most important, we have general ventilation, drying of commercial products, steam and fume removing, some forms of dust removal, induced draught, and ashpit draught, and in fact, for any purpose where air is required to be supplied at atmospheric pressure.

Installing. It was stated in Lesson II that the incoming air approaching the fan had a certain momentum, which is caused by the slight minus pressure existent on the feed side of the wheel.

If the fan is to be fixed into a wall or roof, in such a way that its feed is unimpeded, a maximum efficiency will be obtained. When, however, boxing-in is made necessary, the fan will be seriously handicapped if the box be made too close to it. The less the pitch of the feed edge of the blades, the closer may the box be to the wheel without impeding the feed, but it is not good practice to have any obstruction nearer than the fan wheel's diameter, and a sound rule is to allow a clear distance in an axial direction of 1·2 *times the wheel diameter.* This will be a safe margin for any type of propeller fan.

Peripheral Clearance. In Fig. 9 it was shown that a good percentage of the fan's air was drawn in around the periphery on the feed side of the wheel, and more especially does this apply in the case of the box-bladed type of fan. In practice, therefore, a clearance should be allowed all round the wheel on the feed side of at least equal to a *quarter of the wheel's diameter*.

All conduits leading into the feed box must be of ample area, or else the fan will not be able to fulfil its functions efficiently, even though it be running at such a speed as to overcome slight resistances. To make the total area of all ducts equivalent to the area of the fan is not sufficient, and in such cases restriction of the fan's air volume will occur. It is not possible to set down hard and fast rules as to the sizing of ductwork for propeller fan installations, as each example must be treated on its merits, but the longer the conduit the larger it should be for a given volume in order that frictional losses may be reduced to a minimum. This is the reason why volume fans are unsuited to installations necessitating ductwork unless the latter be very short and of large area. A good general rule is to make the aggregate areas of ducts 25 per cent more than that of the fan.

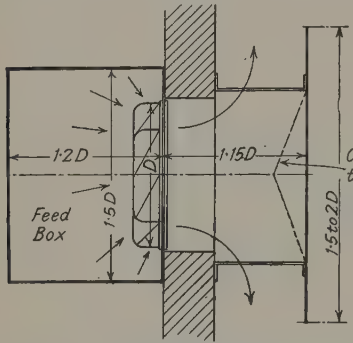

FIG. 12. FEED BOX AND PLAIN BACK-DRAUGHT BAFFLE

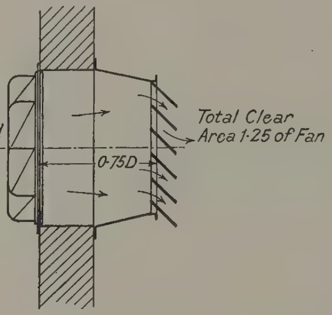

FIG. 13. FIXED LOUVERED SHUTTERS

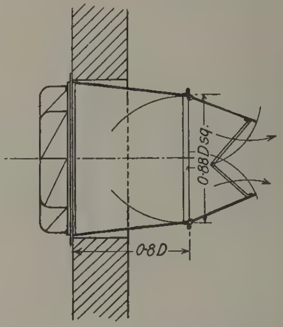

FIG. 14. AUTOMATIC SHUTTERS THAT CLOSE BY THEIR OWN WEIGHT WHEN FAN NOT RUNNING

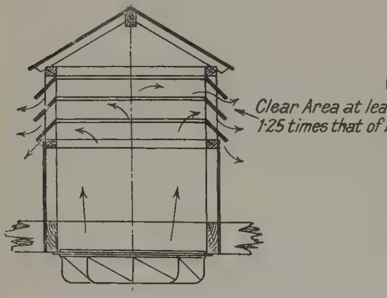

FIG. 15. VERTICAL DISCHARGE WITH LOUVERED TURRET

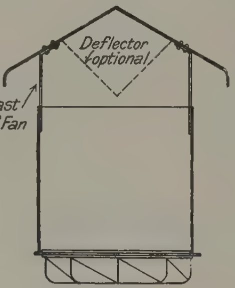

FIG. 16

Some Common Methods of Installing Volume Fans Showing Various Devices for the Prevention of Adverse Back-draughts.

Discharge Clearance. The stream of air on delivery from a fan conforms more or less to a cylinder formation of a diameter equal to that of the fan, and it may be restricted to this course for a short distance without detriment, although any duct employed for this purpose should be slightly greater in diameter than the actual fan wheel. This assists the free escape of the discharged air. A fan should never blow direct into the open atmosphere without the provision of some device capable of arresting back-draughts. It will be obvious that if a fan is blowing a stream of air against an adverse current the effective work performed by that fan will be reduced by the force of the adverse current.

Various means of assuring a maximum discharge of air with a minimum back pressure are shown in the diagrams, Figs. 12, 13, 14, 15, and 16.

Fig. 12 represents a simple baffle plate which can be constructed of iron or timber and is self-explanatory. In this diagram a feed box is shown with the clearance proportions indicated.

Fig. 13 shows a system of louvered shutters which may be fixed permanently open or made to open and close at will.

A system of self-opening and closing shutters is illustrated in Fig. 14, which are so balanced that the force of the discharged air stream is sufficient to maintain them in their open position. When the fan is at rest they close by their own weight.

Figs. 15 and 16 are two examples of roof turrets, the former being of timber and arranged with louvers.

ENGINEERING MATERIALS
TESTING AND PROPERTIES

By R. G. Batson, A.K.C., M.Inst.C.E., M.I.Mech.E.

LESSON VI

HARDNESS AND IMPACT TESTS

Brinell Hardness Testing

This method of hardness testing consists in pressing, under a known load, a hardened steel ball into the material under test and measuring the diameter of the permanent indentation which is produced.

If P = the pressure in kilogrammes,
D = the diameter of the ball in millimetres,
d = the diameter of the indentation in millimetres,

then the hardness number H is equal to the load divided by the spherical area of the indentation,

$$\text{i.e.} \quad H = \frac{P}{1{\cdot}571\, D\,(D - \sqrt{D^2 - d^2})}$$

The value of the hardness number varies with the load applied and with the diameter of the ball used. It has been found that, with homogeneous material, the hardness number is constant provided the ratio $\frac{P}{D^2}$ remains constant. This is due to the fact that indentations, made under such conditions, are similar. The ratios of $\frac{P}{D^2}$ for various classes of materials have been standardized by the B.E.S.A. (See B.E.S.A. specification No. 240, 1926.)

For steels and materials of similar hardness, $\frac{P}{D^2} = 30$, thus, with a 10 mm. ball, $P = 10^2 \times 30 = 3{,}000$ kg. The standard values of $\frac{P}{D^2}$ for other materials are—

For copper alloys and materials of similar hardness . . . . . $\frac{P}{D^2} = 10$

For copper . . . . . . $\frac{P}{D^2} = 5$

For lead, tin, and materials of similar hardness . . . . . $\frac{P}{D^2} = 1$

For thin samples and for other special purposes, balls whose diameter is less than 10 mm. are now being largely used. The diameters of these balls recommended by the B.E.S.A. are 5 mm., 2 mm., and 1 mm.

If a piece of homogeneous material is taken and the hardness numbers obtained by using various values of the ratio $\frac{P}{D^2}$, the values of these hardness numbers when plotted against the corresponding values of $\frac{P}{D^2}$ will give a curve, such as that shown in Fig. 26.

The hardness number increases as $\frac{P}{D^2}$ increases, then becomes approximately constant and finally decreases. It is found, with steel, that the hardness numbers, obtained with $\frac{P}{D^2} = 30$, usually lie on the flat part of the curve. If the value of $\frac{P}{D^2}$ is too high the diameter of the indentation is a large percentage of the ball diameter. Experience has shown that if this percentage is greater than 60 per cent the resulting hardness number often lies on the final part of the curve given in Fig. 26. For this reason the B.E.S.A. state that the diameter of the indentation should not exceed 0·6 of the ball diameter.

The B.E.S.A. further recommend that the thickness of the test specimen shall be at least seven times the depth (t) of the indentation as given by the formula $t = \frac{P}{\pi D H}$, also that the centre of the indentation shall be not less than two and a half times the diameter of the indentation from any edge of the test specimen. These requirements are based on the experimental work of Dr. H. Moore.

The time during which the load is applied to the ball is important, especially with soft materials. This time should not be less than 15 sec. for steel, but should be 30 sec. where the value which is used for $\frac{P}{D^2}$, is 10 or less.

It has been shown by numerous experimenters that for steel an approximate indication of the tensile strength in tons per square inch can be obtained by multiplying the Brinell Hardness number by 0·22. This factor should, however, be used with care as, in some cases, such as cold-worked materials, wide variations occur.

Brinell hardness tests with 10 and 5 mm. balls can be carried out in the universal testing machine by the provision of special holders for the balls. Most makers supply such holders to fit their own type of machine. Where a considerable number of tests have to be carried out, machines specially designed for the work are usually employed. At the present time they are usually of two sizes—

1. Large machines with a maximum load of 3,000 kg. for use with 10 mm. balls (some makers, e.g. Alfred J. Amsler & Co., have designed their machines for a maximum load of 5,000 kg.
2. Machines for use with small balls.

With the latter type of machines the apparatus supplied for measuring the diameter of the indentation must read to an accuracy of ± 0·5 per cent, and special arrangements must be provided to eliminate all trace of vibration or shock. Many machines of the large type are only supplied with measuring microscopes to read to ± 0·05 mm., it is preferable, however, that the accuracy should be ± 0·5 per cent.

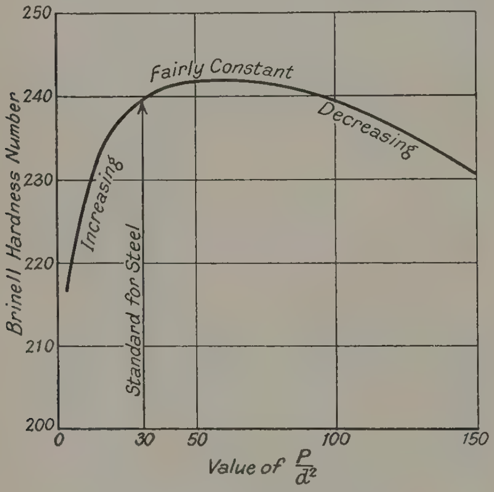

Fig. 26. Hardness Tests

Brinell Hardness Testing Machines. A machine for use with the larger sizes of ball is shown in Fig. 27. This machine is supplied by J. W. Jackman & Co., Ltd., Caxton House, Westminster. The test piece is placed on a table *S*, and is brought into contact with the ball *K* by means of a screw and hand-wheel *r*. The ball is attached to a hydraulic ram to which the pressure is supplied by a small hand pump. The machine has dead weight control of the load applied, this control consists of a small cylinder *a* connected to the top of the press and, therefore, subjected to the same intensity of pressure as the press. A small piston, accurately fitted without packing, works in the small cylinder, and is loaded through a bar *i* by weights *p*. The weights regulate the maximum pressure that can be applied. Any leakage of oil past the piston is collected in a receptacle *d*, from which it is returned to the oil reservoir through the funnel *t*. The valve *v* connects the pressure chamber with the reservoir, and by opening this valve the pressure is released. A pressure gauge *G* is provided to indicate to the operator the rate at which the load is applied.

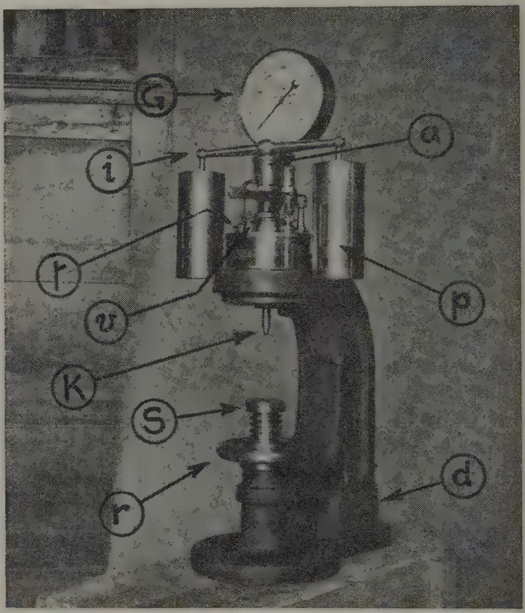

Fig. 27. Brinell Hardness Testing Machine

Machines in which the load is applied by dead weights or a lever system are made by

W. & T. Avery, Ltd., Brown Bayley's Steel Works, Ltd., and Alfred Herbert, Ltd. The machine made by Alfred J. Amsler & Co. employs their well-known pendulum dynamometer.

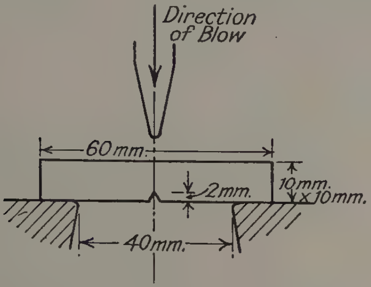

Fig. 28. Beam Type Test Piece

Special machines suitable for small balls are made by Alfred Herbert, Ltd., W. & T. Avery, Ltd., and Vickers, Ltd.

The Brinell test breaks down on hard materials. With commercial balls there is permanent flattening of the ball when materials whose hardness number is greater than 500 are tested. Work-hardened balls can be obtained from Amsler's which are suitable for hardnesses up to 700. The effect of the flattening of the ball is to give a lower hardness number than would be obtained otherwise.

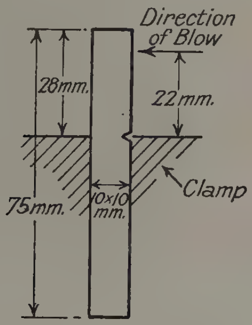

Fig. 29. Cantilever Type Test Piece

Hardness Machines Using Diamond Tools. The difficulty mentioned in the last paragraph is overcome by using diamond tools. A pyramidal diamond is employed in the Vickers, Ltd., diamond hardness testing machine. This machine is a self-contained apparatus which automatically applies the load (from 5 to 100 kg.) to a ball or diamond. The load is applied without inertia for a predetermined time, and is then automatically released. A specially designed microscope of high magnification is attached to the machine, and can be swung over the work to a position immediately over the indentation. The microscope is provided with an indicator giving actual figures from which the diameter of the indentation can be determined.

With a pyramidal or conical diamond the indentations are always similar, so that the same hardness number is obtained whatever the load employed. The Vickers diamond is so designed that the hardness numbers obtained are on the Brinell scale, and true hardness numbers can be obtained outside the limit imposed on the ball method, due to the flattening to which reference has been made previously.

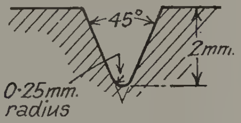

Fig. 30. Form of Notch

The Rockwell direct reading hardness machine (supplied by G. H. Alexander Machinery Ltd., Birmingham) also employs a diamond which in this case is conical with a rounded end. This machine gives a direct reading of the depth of the indentation, and the results are, therefore, not on the Brinell scale. The machine is particularly suitable for rapid determinations of the comparative hardness of materials.

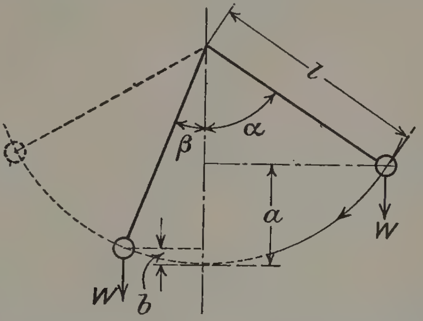

Fig. 31. Method of Calculating Energy Absorbed by Test Piece with Pendulum Machines

Both the Vickers and Rockwell machines are in use at the National Physical Laboratory, Teddington.

NOTCHED BAR TESTING

Notched bar testing was developed on the Continent by Barba, Charpy, Fremont, etc., and in this country by Izod, Seaton, Jude, Stanton, Bairstow, Philpot, etc. Its use was popularized during the war when it was employed on the inspection of aircraft materials. Two methods of carrying out the test are used extensively, and both are standardized by the B.E.S.A. These are shown diagrammatically in Figs. 28 and 29. In the first method (Fig. 28) a notched test piece, 60 mm. long, is placed on supports 40 mm. apart, and is struck at the centre opposite the notch by the tup of a hammer, and the energy absorbed from the hammer in breaking the test piece is measured.

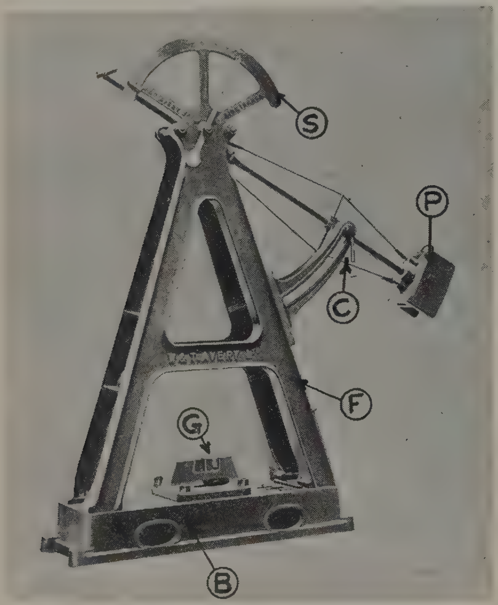

Fig. 32. Notched Bar Testing Machine
Cantilever (Izod) Method

In the second method (Fig. 29) the test piece is gripped in a vice, and broken as a cantilever by a blow applied 22 mm. above the vice. In both cases the test piece is 10 × 10 mm. in cross-section with a vee notch having an angle of 45° and a root radius of 0·25 mm. (See Fig. 30).

The same type of machine is usually employed in both cases. It consists of a weighted pendulum having its centre of percussion at the point at which the test piece is struck.

The method of calculation is shown by Fig. 31.

The energy in the hammer before test

$$= Wa = Wl(1 - \cos \alpha)$$

The energy in the hammer after test

$$= Wb = Wl\,(1 - \cos \beta)$$

Reduction in the energy in the hammer and therefore the energy taken in breaking the test piece

$$= Wa - Wb = Wl\,(\cos \beta - \cos \alpha)$$

where

Wl = The moment of the pendulum.

α = The angle of the pendulum before the test.

β = The angle of the pendulum after the test.

The machine common in this country is the Izod machine, manufactured by W. & T. Avery, Ltd. The original machine of this type had an initial striking energy of 23 ft.-lb., and employed a test piece $\frac{3}{8}$ in. × $\frac{3}{16}$ in. in cross-section. The same type of machine was employed for the larger size of test piece (10 mm. × 10 mm.), developed during the war, but with the energy increased to 120 ft.-lb. and the framework strengthened. This machine is illustrated in Fig. 32. The heavy base *B* has the grips *G* and an **A** frame *F* bolted to it. The top of the frame *F* supports the axis of the pendulum *P*, which revolves on ball bearings. An extension of the pendulum works a pointer over a scale *S*, which is graduated directly to read in ft.-lb. The pendulum is released by an adjustable catch *C*, attached to a bracket on the side of the **A** frame.

POWER TRANSMISSION BY BELTING

By W. G. Dunkley, B.Sc.

LESSON VII

COEFFICIENT OF FRICTION—BELT TO TRANSMIT GIVEN HORSE-POWER

Coefficient of Friction. We have seen earlier that the value chosen for μ, the coefficient of friction, makes a considerable difference in the slipping ratio $\frac{T}{t}$. Before we work some examples we must, therefore, decide what is a reasonable value to allow for μ in our calculations. Mr. Wilfred Lewis made some extensive tests on this important question. He found that μ had values varying from 0·25 to 1·00 according as the conditions varied. The value of μ increased as the slip of the belt on the pulley increased. There must necessarily be some slip of the belt over the pulley, that is, the speed of the belt is not equal to that of the pulley surface, and some rubbing or relative motion occurs. Excessive slipping, due to overload, is, however, injurious to the belt, since it dries up the surface. A slip of 2 per cent is recommended as reasonable, and with this percentage slip values of μ varying from 0·44 to 0·9 were found.

The value of μ is not under our control, except in so far as the belt is kept in good condition. Mr. Lewis states that a value of $\mu = 1{\cdot}00$ can be maintained under good working conditions if the belt is kept in good condition. If we assume a value of $\mu = 0{\cdot}40$ we shall be using a reasonable value. We shall, therefore, adopt this value in all the following calculations. For driving belts which have to be repeatedly moved across the face of the pulley, as in planing machine drives, it might be well to reduce the value to 0·30. We shall now work a number of examples.

Effective Driving Load. We have seen previously that the torque produced by the belt is given by multiplying the difference in the tensions in the two sides of the belt by the radius of the pulley. This difference in the tensions or $(T - t)$ we shall refer to as the *effective driving load.*

Horse-power Formulae.

Let P = effective driving load in pounds.
D = diameter of pulley in inches.
N = revolutions per minute of pulley.
V = velocity of belt in feet per minute $= \pi \times D \times N \div 12$.

$$\begin{aligned}\text{Horse-power} &= P \times V \div 33{,}000\\ &= \frac{P \times \pi \times D \times N}{12 \times 33{,}000}\\ &= \frac{P \times D \times N}{126{,}000} \qquad (4)\end{aligned}$$

Examples on Horse-power Transmitted

Example 10. A single belt 5 in. wide runs on a pulley 36 in. diameter, making 200 rev. per minute. Angle of lap 175°, $\mu = 0{\cdot}4$.

Find: The horse-power which can be transmitted and the belt tensions.

We must first find the permissible ratio of $\frac{T}{t}$.

$$\begin{aligned}\text{Log. } \frac{T}{t} &= 0{\cdot}007578 \times \mu \times a\\ &= 0{\cdot}007578 \times 0{\cdot}4 \times 175\\ \frac{T}{t} &= 3{\cdot}4 \quad \therefore \quad T = 3{\cdot}4 \times t.\end{aligned}$$

Now in order to find the horse-power we must know the value of P.

Now $P =$ effective driving load $= T - t$

$$\begin{aligned}&= 3{\cdot}4\,t - t = (3{\cdot}4 - 1)t\\ &= 2{\cdot}4t.\end{aligned}$$

$$\therefore \frac{P}{T} = \frac{2{\cdot}4t}{3{\cdot}4t} = \frac{2{\cdot}4}{3{\cdot}4}$$

$$P = \frac{2{\cdot}4}{3{\cdot}4} \times T.$$

We have found the relation between P and T, because we must determine our value of P such that the maximum stress in the belt is not above the reasonable value permissible. For a single belt, assuming it to be laced, we decided in Lesson VI to use a stress of 320 lb. per sq. in.

The area of our belt assuming it to be $\frac{3}{16}$ in. thick is $5 \times \frac{3}{16}$. The permissible value of T, which gives a stress of 320 lb. sq. in., is, therefore, given by

$$T = 320 \times 5 \times \tfrac{3}{16} = 300 \text{ lb.}$$

$$\text{Then } P = \frac{2\cdot4}{3\cdot4} \times T = \frac{2\cdot4}{3\cdot4} \times 300 = 212 \text{ lb.}$$

Then

$$\text{h.p.} = \frac{P \times D \times N}{126{,}000} = \frac{212 \times 36 \times 200}{126{,}000} = \underline{12}$$

$$P = T - t \quad \therefore \quad t = T - P = 300 - 212 = \underline{88 \text{ lb.}}$$

What would be the minimum initial belt tension t_1?

$$t_1 = \frac{T + t}{2} = \frac{300 + 88}{2} = \underline{194 \text{ lb.}}$$

Order of Procedure. We can arrange the order of procedure thus—

1. Find $\frac{T}{t}$.
2. Now $P = T - t$

$$= \left(\frac{T}{t} - 1\right)t = \frac{\left(\frac{T}{t} - 1\right)t \times \frac{T}{t}}{\frac{T}{t}} = \frac{\left(\frac{T}{t} - 1\right)}{\frac{T}{t}} \times T \qquad . \qquad . \qquad (5)$$

3. Decide on the allowable value for T.
4. Substitute the value of T in (5) and thus find P.
5. Find the horse-power from (4).

EXAMPLE 11. A single belt, 5 in. wide, runs on a pulley 12 in. diameter making 600 rev. per minute. Angle of lap 130 degrees. $\mu = 0{\cdot}4$. Find the horse-power which can be transmitted and the belt tensions. In this example we have the same belt speed as in Example 10, the only change is in the angle of lap. We will now follow our order of procedure in a systematic way.

1. $\text{Log. } \frac{T}{t} = 0{\cdot}007578 \times 0{\cdot}4 \times 130$

$$\text{Then } \frac{T}{t} = 2{\cdot}47.$$

2. $$P = \frac{\left(\frac{T}{t} - 1\right)}{\frac{T}{t}} \times T. = \frac{(2{\cdot}47 - 1)}{2{\cdot}47} \times T. = \frac{1{\cdot}47}{2{\cdot}47} \times T.$$

3. T = allowable stress per square inch × area of belt in sq. in.

$$\therefore \quad T = 320 \times 5 \times \tfrac{3}{16} = 300, \text{ as before.}$$

4. $$P = \frac{1{\cdot}47}{2{\cdot}47} \times T = \frac{1{\cdot}47}{2{\cdot}47} \times 300 = \underline{178 \text{ lb.}}$$

5. $$\text{h.p.} = \frac{P \times D \times N}{126{,}000} = \frac{178 \times 12 \times 600}{126{,}000} = \underline{10{\cdot}2.}$$

We see the change that the angle of lap has made.

$$t = T - P = 300 - 178 = 122.$$

Initial belt tension must not be less than

$$\frac{T \times t}{2} = \frac{300 + 122}{2} = 211 \text{ lb.}$$

EXAMPLES. Following the above procedure, work out the following examples—

(*a*) Pulley 24 in. diameter, r.p.m., 320. Single belt 4 in. wide, cemented joint. Angle of lap 180°. $\mu = 0{\cdot}4$ Find h.p. *Ans.* 12·5.

(*b*) Find h.p. for a double belt 4 in. thick. Cemented joint. *Ans.* 18·3.

(*c*) Change the angle of lap to 140°.

Ans. Single belt. h.p. = 10·9. Double belt, 15·9.

PATENTS FOR INVENTION

By W. E. Dommett, Wh.Ex., M.I.Mar.E., A.F.R.Ae.S.

LESSON VI

OFFICIAL PROCEDURE

When the specification is filed at the Patent Office it is referred to an examiner, whose duty it is to see that the requirements of the Patents Acts and Rules have been complied with. It is also his duty to make an investigation for the purpose of ascertaining whether the invention claimed has been wholly or in part claimed or described in any specification published during the last fifty years. The examiner's report is communicated to the applicant and, if there are any objections raised, the applicant has the opportunity then of amending his specification to overcome these objections. In so far as the examiner raises objections on purely formal matters the instructions already given in preceding sections should be followed; for example, supposing the title is objected to as being a fancy title, the form of amendment should now be clear. The actual amendment itself can be effected by crossing out, cancelling the title, typing or writing in the new title, and initialing the amendment at the left-hand side. It will be gathered from this that an amendment does not necessarily call for the provision of new pages.

If the objection is against the clearness of a certain part of the specification, then the paragraph concerned should be recast in such a way as to overcome the objection and, if this re-arranged paragraph cannot be conveniently put into the specification without upsetting the sequence of paragraphs, the difficulty can be overcome by retyping the page or pages of the specification so affected so that the specification reads as a complete continuous document. When such a re-arrangement is made, duplicates of the freshly typed pages should be returned with the amended specification, and the superseded pages should be cancelled, initialed, and also returned.

Generally speaking, the most difficult objection to meet is that arising from the finding of prior specifications as a result of the examiner's investigation. The printed copies of the cases concerned can be obtained from the Patent Office, occasionally they can be found in the public libraries of the larger towns, and sometimes brief descriptions, known as abridgments, can be looked up in copies of the *I.O.J.* (*Illustrated Official Journal of Patents*), which is issued week by week to a greater number of libraries.

I think it would be wise to keep to our imaginary invention by way of example and to suppose that, as a result of the investigation thereon, a prior specification is cited. Having obtained a copy of this, we find that it relates to a fuel pump having a lever between the plunger and cam, and the fulcrum pin of the lever is eccentrically mounted on a disc adjustably fixed to the lever. It therefore follows that our Claim 1 for all constructions of a lever with an adjustable fulcrum cannot be sustained since one construction has already been proposed, and that therefore we must narrow our claim. Looking into our own specification we find that the distinguishing feature is that the fulcrum is supported in a block, slidably mounted in a guide or guides, which is the subject of our Claim 2. As far as the claims are concerned, Claim 1 would be cancelled, this being done by crossing it out and initialing it and re-numbering the remaining claims in sequence. The statement of invention (fifth clause) must be similarly amended to agree with this new claim, and some acknowledgment should be inserted by way of reference to this prior case. If only one case is cited then the best acknowledgment would consist in putting in a paragraph before the preamble to the claims, stating that: "It has previously been proposed to provide the operating lever of a fuel pump with an eccentrically mounted fulcrum."

On the contrary, should there be a number of cited cases, it may be argued that the general provision of variable fulcrumed levers is so common that it practically forms a sub-type of pump. In this case the acknowledgment is best effected by expanding the opening clause which at present reads: "My invention relates to fuel pumps for internal combustion engines, and more particularly the variable stroke type of pump," by adding some such phrase as: "Wherein a lever having an adjustable fulcrum is placed between the plunger and driving cam."

If the amendment of specification is carried out on these lines there is very little risk of further objections being raised. It is important to notice that no amendment should be made by pasting a slip of paper over cancelled parts of the original documents and, what is more important still, no amendment should be made beyond that called for by the official letter.

A period of two months is allowed in which an amendment must be effected, but, if necessary, an extension of time can be obtained by applying on Form 7, which requires a fee of 10s. for each month's extension. If the amendments made are not satisfactory, the Comptroller appoints a time for hearing the inventor's and the examiner's views, and makes a decision as to how the specification should be dealt with.

At this stage it would be as well to deal with the important question of disconformity. When the invention claimed in a complete specification is not substantially the same as that described in the provisional specification there is said to be disconformity between the two, and the applicant may have to agree to the cancelling of the provisional specification, and post-dating the application to the date of filing the complete specification. To explain this by reference to our imaginary invention, suppose that in the provisional specification we had stopped at the fifth clause referring to any adjustable fulcrumed lever and had not put in any of the following clauses; then as soon as we cut out the broad Claim 1 of our complete specification on amendment and limited the invention to the specific sliding block construction, we get disconformity because the invention as now set out was not foreshadowed in the provisional specification.

When all the official requirements have been met, and the complete specification is in order, it is accepted, and its acceptance is advertised in the *I.O.J.* and later the specification is printed. The specification ordinarily should be accepted within fifteen months of the date of application, but three months' extension may be obtained in exceptional circumstances. Even though the specification is ready for acceptance a request for its delayance is generally agreed to up to the period of fifteen months, the usual reason for asking for the delay is that the applicant is contemplating making applications for patents in foreign countries.

When the specification is advertised as accepted, it is open to opposition, e.g. somebody may come along and oppose the grant of the patent on the ground that the invention has been stolen and, should such opposition proceeding arise on this or any other recognized grounds of opposition, then a patent agent should undoubtedly be consulted. If there is no opposition then the Comptroller will seal and issue the patent at the request of the inventor. This request is made on Patents Form 12, which requires £1 stamp.

Provided that the application takes an average course it will be observed that the official fees total £5, £1 on application Form 1, £3 on complete specification Form 3, and £1 on sealing Form 12. This expenditure of £5 covers the whole period for four years from the date of application. In the fifth year a renewal fee of £5, stamped on Patent Form 14 is required, £6 for the sixth year, £7 for the seventh, and so on to the full life of the patent, i.e. 16 years.

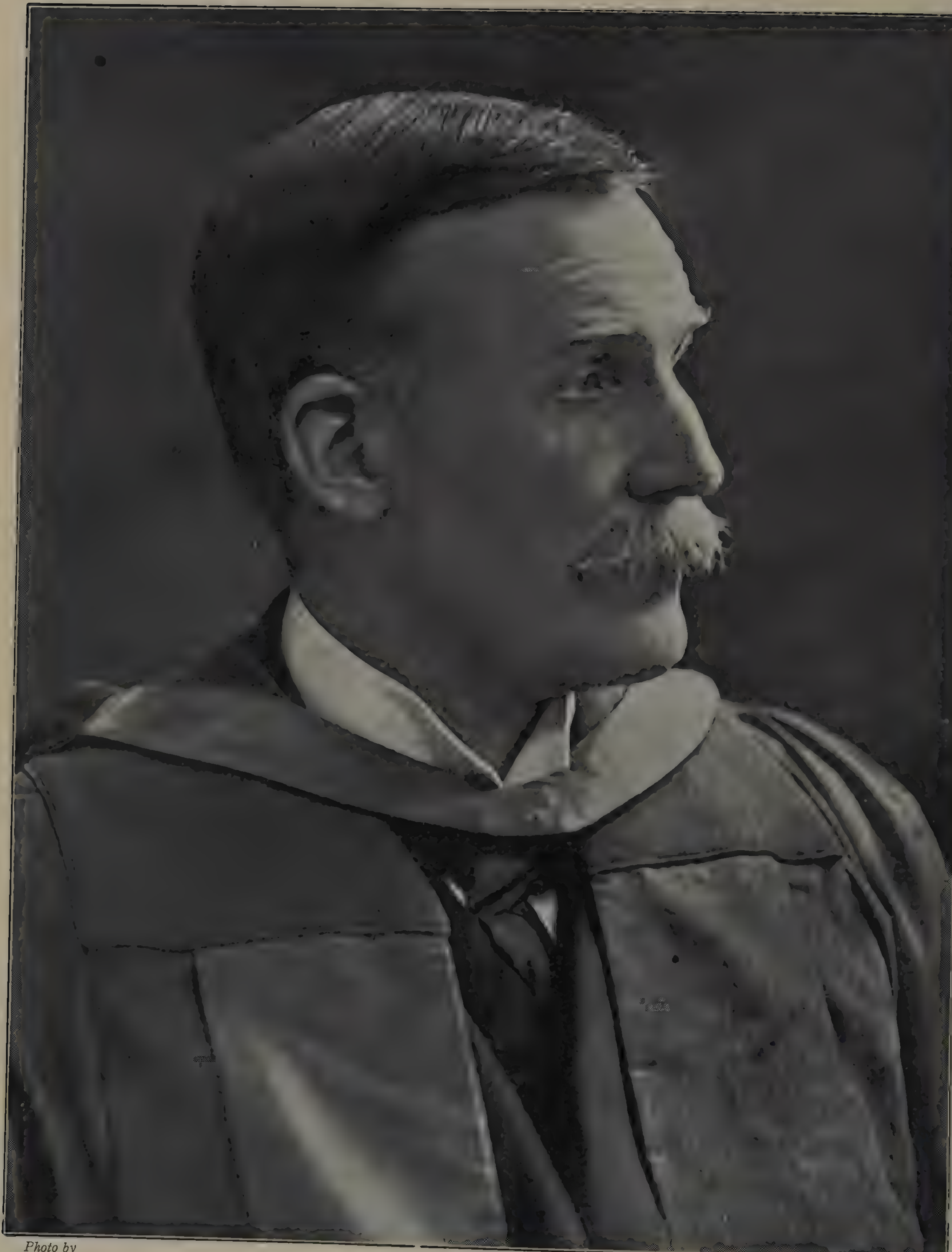

Photo by *Lafayette*

SIR ARCHIBALD DENNY, BART., LL.D.

(5462)

THE BRITISH ENGINEERING STANDARDS ASSOCIATION

By Sir Archibald Denny, Bart., LL.D.
Chairman, B.E.S.A. Main Committee

This Association was founded as The British Engineering Standards Committee in 1901, by the Institutions of Civil Engineers, Mechanical Engineers, Naval Architects, Electrical Engineers, and The Iron and Steel Institute.

The steel makers of the Kingdom found that they were put to enormous expense in having to keep hundreds of sets of rolls for steel sectional materials, and moved for standardization. As a result of the Standards Committee's work, a total of 175 different sections of angles, equal and unequal, bulb angles, bulb tees, etc., was published in 1903, which not only greatly reduced the number of rolls, but also improved the sections themselves in design. In 1918, at a revision, this number was further cut down to 115, and it was found by a census of production taken in 1913, the last pre-war year, that nearly 95 per cent of the total sections rolled were to standard.

That first step was so successful that tramway rails were next tackled. It was found that there were 75 different sections in use, involving a corresponding number of sets of rolls. As a result of conferences and meetings between users and makers, this was reduced to one standard profile and only four sizes.

The above two specifications led to an enormous saving, stated at the time by the steel makers, to be not less than 5s. per ton rolled, and some millions of tons were rolled each year. This led to others applying for similar help, and Portland cement, for which there were dozens of different specifications, was dealt with at the request of the makers. Agreement between them and the leading consulting engineers and the Government departments was reached, with the result that a National Standard was set up and published, and a harassed industry was put on a firm foundation.

Standardization as carried out by the Association is in no sense fossilization, and that specification has been frequently revised, the latest revision being dated 1925. At these revisions the new specification always called for higher tests, and British Portland cement takes a leading place in the world.

Before the war the Association made steady though not very rapid progress, but during the war its activities, at the request of the Government, then practically the sole purchaser, were much and rapidly extended.

As national propaganda, all the leading specifications were translated into French, Italian, Spanish, and Portuguese. Local committees were founded, not only in our own Dominions, but also in many of the South American Republics, where the translated specifications were very welcome.

In 1918 the committee was incorporated under its present name of B.E.S.A., and since then the Institution of Automobile Engineers and the Royal Institute of British Architects have joined the Founder Institutions in nominating the main committee.

The method of working is as follows: The main committee controls the general policy, but the specifications are actually made by special committees appointed by the main committee.

These sectional committees, it is laid down, must be fully representative of makers and users, and generally some experts and independent consulting engineers, interested in the particular piece of work under consideration, are also appointed.

The Association does not of itself initiate standards, and only appoints committees when requested to do so by responsible bodies of makers or users or by both, or by Government departments. Before appointment, conferences are always held to discover whether there is a real need and demand by both makers and users. Since the Armistice, no fewer than 415 specifications, new or revised, have been published, so that merely to give a list of them would take more space than I have at my disposal; but there are working on them from time to time, over 2,000 engineers and business men on over 400 committees, sub-committees, and panels, and these gentlemen work without fees, neither are their expenses paid—a tribute to the value they place on the work.

These gentlemen, and representatives of various trade associations, constituted until recently the membership of the Association.

To give another and very recent example of the work of the B.E.S.A., the Mining Association of Great Britain invoked its assistance with a view to cheapening their pit materials. An important specification on pit rails has just been completed, reducing the number of flat-bottomed rail sections from 25 to 9, and another most useful one for steel arches for use in mines has just been issued, and work is proceeding on many other items.

As already mentioned, standardization does not lead to fossilization as the specifications are frequently revised.

Perhaps the word "standard" does not give a quite correct definition of the work. Generally no theoretically perfect standard is aimed at; rather is the work "simplification" and the elimination of redundant patterns.

Starting in 1901 for the purpose of dealing with steel sectional materials, cement, sand-lime bricks, and many other such building materials have been standardized. But even materials which, at first sight, might not be considered covered by the term "engineering" have been dealt with, such as fabrics for the wings of aeroplanes. Oils have also been dealt with in specifications in this country both for fuel and insulating oils, while in South Africa a specification dealing with "standard" coal has been published.

The question may be asked, why not leave each trade to do its own standardizing work? One answer is that they could not do it so economically because, instead of having the immediate advantage of an experienced and fully trained staff, they would each require to create a separate and nearly as numerous a staff. But the principal advantage follows from the fact that no trade or business is really independent, each profession, trade, or industry depending more or less on every other, and thus the advantage of a central body is that each trade is at once in touch with all others through the Association.

Recently, a new class of membership has been created; any individual interested may apply for membership at an annual subscription of £2 2s., which brings him many privileges and gives him the right to representation on the main committee, through an advisory committee of all the members.

There is no doubt as to the value of the work, and this, perhaps, is best seen by noting that 17 other countries have copied this country and have established similar National Associations, most of them since the war. Many of them have applied to 28 Victoria Street for advice and assistance, and it has never been refused to them.

MATHEMATICS FOR ENGINEERS

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LESSON XV

TRIGONOMETRY

Tangent of an Angle. Suppose it is necessary to find the height of a chimney or a tower, or any such high, inaccessible object, how are we to set about it? One method is to measure the length of the shadow on a sunny day. This alone is not sufficient, for the shadow changes its length from hour to hour, as the sun gets higher or lower. By means of an upright stick, of known height, whose shadow is also measured at the same time, we can find the unknown height of the chimney by simple proportion. Chimney and stick are represented in Fig. 50, *AB*, and *ab* respectively; *BC* and *bc* are shadows, of lengths 160 ft. and 8 ft. respectively; and the height of the stick is 5 ft. For this time of day, when a 5 ft. stick casts an 8 ft. shadow, height and shadow are always in the proportion of 5 to 8; the height, then, is $\frac{5}{8}$ of the shadow length. For the chimney,

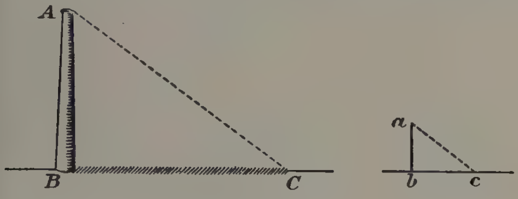


FIG. 50

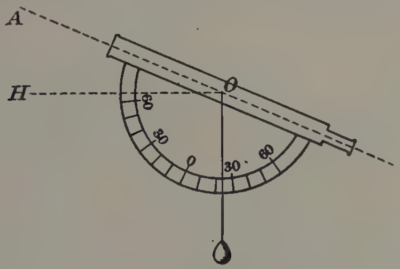


FIG. 51

$$h = \frac{5}{8} \times 160 = 100 \text{ ft.}$$

We see in this instance that there is, for any altitude of the sun, a factor ($\frac{5}{8}$ here) which will convert shadow length into height—or, in geometrical language for the triangle *ABC*, will convert base into height. The factor evidently varies with the altitude of the sun, that is, with the angle *ACB* (or *acb*, which is equal to *ACB*). Each "angle of elevation" of the sun will have its own conversion factor, and in practice it is the angle which is measured, by means of a theodolite, an instrument consisting essentially of a telescope which is pivoted on a horizontal axis, and to which is attached a vertical graduated circle. A primitive form is sketched in Fig. 51. The conversion factor is then found from tables; it is there called the **tangent** of the angle (*natural* tangent, its logarithm being often given in another table of *logarithmic* tangents, to save the trouble of referring to two tables when logarithms are used in the calculation).

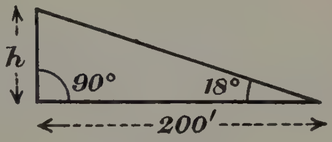


FIG. 52

As an example of its use, we will solve the following problem. At a distance of 200 ft. from a tower, the angle of elevation of the top is 18°. What is the height of the tower? To find the height, we have to multiply the base (200 ft.) by the "conversion factor" for 18°, i.e. by the "tangent of 18°" (usually written shortly as *tan* 18°). Thus (see the diagram in Fig. 52)—

$$\begin{aligned} h &= 200 \tan 18^\circ = 200 \times 0{\cdot}3249 \\ &= \underline{64{\cdot}98 \text{ ft.}} \end{aligned}$$

The value 0·3249, of *tan* 18°, was obtained from tables.

It should be clear from this example that we have here the germ of a method of calculating sides and angles of right-angled triangles, and, since *any* triangle can be divided into two right-angled triangles, of performing calculations upon the sides and angles of any triangle.

The name for this branch of mathematics is Trigonometry, a word derived from two Greek words meaning *triangle* and *measure*. This and a few succeeding lessons will be devoted to those portions of trigonometry that are needed in engineering applications.

Referring to a right-angled triangle (see Fig. 53), calling the base b, the perpendicular p, and the angle A, the rule we have used above can be expressed by the formula

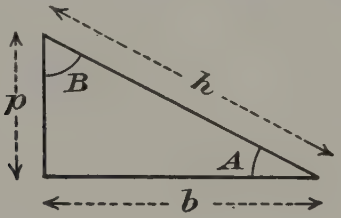
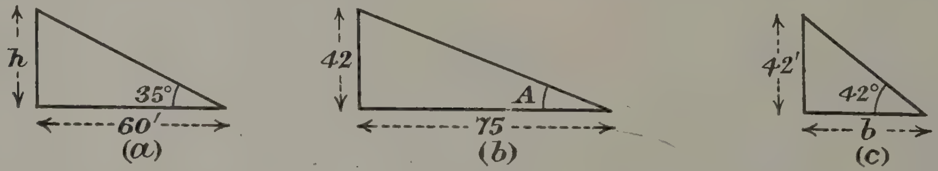


FIG. 53

$$\mathrm{p = b \tan A} \qquad . \quad . \quad (A)$$

In becoming familiar with this formula the student must be prepared to find many instances in which the triangle is turned round, so that the "base" will not always be at the bottom. It is therefore important to notice the *relative* positions of the two sides and the angle. The two sides include the right angle (we shall see how to deal with the hypotenuse also, by the use of other "conversion factors," later), and the angle is *opposite* one of these sides, and *next to* the other. Now notice that the side *opposite* the angle is on the *opposite* side of the = sign in the formula, while the side *next to* the angle is on the *same side* of the = sign in the formula. If this relation between triangle and formula is kept in mind, there should never be any doubt as to which angle to use to find any side—go for the angle *opposite* the required side.



(*a*) (*b*) (*c*)

FIG. 54

EXAMPLES. 1. When the angle of elevation of the sun is 35°, the shadow of a chimney is 60 ft. long. Find the height of the chimney.

The facts are diagrammatically represented in Fig. 54 (*a*). Here the unknown side is opposite the angle 35°. So

$$\begin{aligned} h &= 60 \tan 35° = 60 \times 0{\cdot}7002 \\ &= \underline{42{\cdot}012 \text{ ft.}} \text{ (say, 42 ft.).} \end{aligned}$$

2. What is the angle of elevation of the sun when the shadow of the above tower is 75 ft. long?

The facts are represented in Fig. 54 (*b*), and here, knowing the two sides, including the right angle, we need the angle A. Using the fundamental formula,

$$42 = 75 \tan A$$

whence $$\tan A = \frac{42}{75} = 0{\cdot}56$$

Now, looking in tables for an angle whose tangent is 0·56, we find it is just over 29°. Using the difference columns, we get, to the nearest minute,

$$\underline{A = 29° \, 15'}$$

As we shall often be finding angles, it may be useful to put the formula in the simplest form for use in such cases. Dividing by b, we find

$$\mathrm{\tan A = \frac{p}{b}} \qquad . \quad . \quad (B)$$

It is worth while remembering (B) as well as (A).

3. What will be the length of the shadow when the angle of elevation of the sun is 42°?

The facts are represented in Fig. 54 (*c*). Here we are not given the angle *opposite* to the side, b, that we require; it is easily found, however, by subtracting from 90°; the angle opposite b is 48°. Thus,

$$\begin{aligned} b &= 42 \tan 48° = 42 \times 1{\cdot}1106 \\ &= \underline{46{\cdot}65 \text{ ft. (approx.).}} \end{aligned}$$

4. The angles at the base of an isosceles triangle are each 55°, and the base is 6 in. Find its height and area.

The triangle is sketched in Fig. 55, and divided by a perpendicular into two equal right-angled triangles. From one of these, we have

$$h = 3 \tan 55^\circ = 3 \times 1{\cdot}4282$$
$$= \underline{4{\cdot}2846 \text{ in.}}$$

Therefore

$$\text{area} = \tfrac{1}{2} \times 6 \times 4{\cdot}2846$$
$$= \underline{12{\cdot}8538 \text{ sq. in.}}$$

(Four figure accuracy, say, 12·85 sq. in., is all that is really reliable, or desirable.)

Returning to the height problem, it will often be impossible to measure the distance of the point of observation from the foot of the chimney, on account of surrounding buildings, and when the height of, say, a mountain is required, this will be quite impossible. If we take *two* observations, from two points on the level, in line with the chimney, and measure their distance apart, we shall have enough to construct a scale diagram, and whenever it is possible to construct a scale diagram, it is always possible to work out the result—and that more accurately—by calculation.

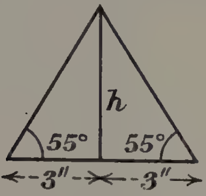


FIG. 55

EXAMPLE 5. From points 120 ft. apart, the angles of elevation of the top of a chimney are 17° and 31°, the points being in line with the chimney, and on the same side of it. Find the height of the chimney.

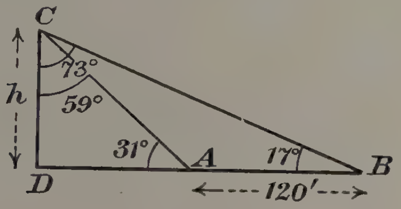


FIG. 56

The data are represented diagrammatically in Fig. 56, A and B being the two points, and CD the vertical through the top of the chimney. We have to determine h, and to do this we must express the known length, AB, in terms of h. In the right-angled triangle, CAD, $AD = h \tan 59^\circ$ (the angle *opposite* AD is $90^\circ - 31^\circ = 59^\circ$), and similarly in the triangle CBD, $BD = h \tan 73^\circ$ $(90^\circ - 17^\circ = 73^\circ)$. But

$$AB = BD - AD$$
$$120 = h \tan 73^\circ - h \tan 59^\circ$$
$$= 3{\cdot}2708\, h - 1{\cdot}6643\, h \text{ (from tables)}$$
$$= 1{\cdot}6065\, h$$

$$\text{Hence } h = \frac{120}{1{\cdot}6065} = \underline{74{\cdot}7 \text{ ft. (approx.)}}$$

6. The base of a triangle is 6 in. long, and the angles at its ends 35° and 45°. Find its height and area.

In Fig. 57

$$AB = AD + DB$$
$$6 = h \tan 55^\circ + h \tan 45^\circ$$
$$= 1{\cdot}4281h + 1{\cdot}0000h$$
$$= 2{\cdot}4281\, h$$

$$\text{Whence } h = \frac{6}{2{\cdot}4281} = \underline{2{\cdot}47 \text{ in. (approx.)}}$$

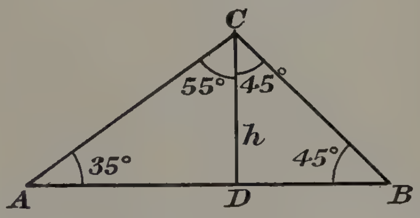


FIG. 57

Using this value of h, we have,

$$\text{area} = \tfrac{1}{2} \times 6 \times 2{\cdot}47 = \underline{7{\cdot}41 \text{ sq. in.}}$$

EXERCISE No. 25

1. At 50 yd. from a house, the angle of elevation of the housetop is 15°. Find its height.
2. Find what the angles of elevation would be at 100 yd. and at 25 yd.
3. At what distances would the angles of elevation be 30°, 5°?
4. The diagonal of a rectangle makes an angle of 35° with the longer side. The shorter side is 7·5 in. Find the longer side and the area.
5. A tower is surmounted by a flagstaff. From a point on the ground, 80 ft. from the tower, the angle of elevation of the top of the tower is 51°, and that of the top of the staff 57°. How long is the staff?
6. From a window, 20 ft. above the ground, the angle of depression of the foot of a chimney is 12°, and the angle of elevation of the top 35°. How far away, and how high, is the chimney?
7. A boat is sailing towards a cliff. The angle of elevation of the top is 8°; 200 yd. nearer in, the angle is 13°. How high is the cliff?
8. From the ends of a viaduct, the angles of depression of a point in the valley between are 18° and 29°. The viaduct is 200 yd. long. How high is it?

9. On top of a hill is a flagstaff, 42 ft. high. From where I stand, the angle of elevation of the top of the staff is 12°, and that of the bottom 10°. How far away (horizontally) is the staff, and how much higher is the hill top than the point where I stand?

Sine and Cosine. We next proceed to show how problems in which we have to deal with the hypotenuse of a right-angled triangle may be solved by a similar method. We need *two* conversion factors, one to convert *h* into *p* (Fig. 53), and the other to convert *h* into *b*. The one for the first case is called the sine of the angle *A* (written *sin A*), so that

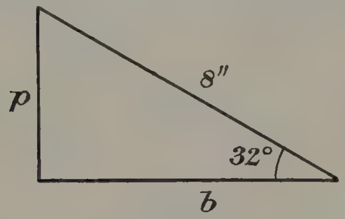


FIG. 58

$$p = h \sin A \qquad . \quad . \quad . \quad (C)$$

Note that in (*C*) the *opposite* and *next* relation between sides and angle holds, as it did for (*A*) above. The other conversion factor is called the cosine of the angle *A* (written *cos A*), so

$$b = h \cos A \qquad . \quad . \quad . \quad (D)$$

In (*D*), since the two sides include the angle, the *opposite* rule will no longer apply.

EXAMPLE 1. The hypotenuse of a right-angled triangle is 8 in. long, and one angle is 32°. Find the other two sides.

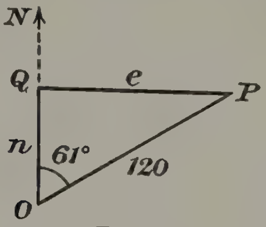


FIG. 59

The data are represented in Fig. 58. From it,

$$\begin{aligned} p &= 8 \sin 32^\circ = 8 \times 0{\cdot}5299 \\ &= \underline{4{\cdot}24 \text{ in. (approx.)}} \\ B &= 8 \cos 32^\circ = 8 \times 0{\cdot}8480 \\ &= \underline{6{\cdot}784 \text{ in. (approx.)}} \end{aligned}$$

2. A ship sails 120 miles in a direction 61° E. of N. Find its northing and easting (i.e. the distances to N. and E. of its starting point).

In Fig. 59, *OP* is the course, *ON* the line due N. through *O*, and *PQ* perpendicular to *ON*. We have to find the northing, *OQ* or *n*, and the easting, *QP* or *e*.

$$\begin{aligned} n &= 120 \cos 61^\circ = 120 \times 0{\cdot}4848 \\ &= \underline{58{\cdot}18 \text{ miles (approx.)}} \\ e &= 120 \sin 61^\circ = 120 \times 0{\cdot}8746 \\ &= \underline{104{\cdot}95 \text{ miles (approx.)}} \end{aligned}$$

This example is important as showing the use of *sin* and *cos* as *resolving* factors; we have found the *resolved parts* or *components* of the displacement *OP* in the N. and E. directions. We shall use sines and cosines for this purpose very often, in dealing with forces and displacements.

3. A load of 2 tons is being drawn up an inclined plane by a rope parallel to the plane, the inclination being 20°. Neglecting friction, find the tension of the rope.

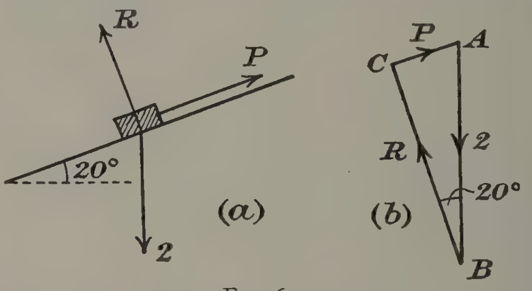


FIG. 60

The load and the forces acting on it are represented in Fig. 60 (*a*), the weight 2 tons vertically, the pull *P* parallel to the plane, and the reaction of the plane *R*, perpendicular to the plane. The triangle of forces for these is *ABC*, where *AB* represents 2 tons vertical, and *CA* and *CB* are parallel to *P* and *R* respectively, and consequently represent them. In *ABC*, $\angle B = 20^\circ$, and $\angle C = 90^\circ$.

$$\begin{aligned} \text{Thus, } P &= 2 \sin 20^\circ = 2 \times 0{\cdot}3420 \\ &= \underline{0{\cdot}684 \text{ tons.}} \\ R &= 2 \cos 20^\circ = 2 \times 0{\cdot}9397 \\ &= \underline{1{\cdot}8794 \text{ tons.}} \end{aligned}$$

Sines and cosines will be further treated in the next Lesson.

ANSWERS TO EXERCISE NO. 24

1. 0·968 in.
2. $7\frac{11}{16}$ ft., 154·6°, 20·74 ft.
3. 0·0403 sq. in.
4. 0·7854, 1·0472, and 1·4137 radians; 57° 18′, 17° 12′, 68° 28′.
5. 723·1.
6. 13·61 and 314·2 rad./sec.

APPLIED MECHANICS

By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XVI

GRAPHIC STATICS

There are many engineering problems whose solution is best effected by graphical methods, these frequently being less tedious, more easily understood, and, when skilfully used, providing solutions of a high order of accuracy. Skill and accuracy in drawing and measurement, however, are indispensable, and such practice as is necessary to acquire these must be undertaken by those using graphical methods.

We are already aware that a force, which is a vector quantity, can be represented completely by a straight line, and this enables us to deal with our forces by the simple graphical methods now to be explained.

Triangle of Forces

The theorem of the *triangle of forces* states that, if three forces acting at a point are in equilibrium, then these three forces can be represented in magnitude and direction by the three sides of a triangle taken in order, the sides being drawn parallel to the directions of the forces which they represent.

Suppose the three forces, AB, BC, and CA, to act upon a body and, in so doing, produce equilibrium, that is, the body remains at rest. Choose any convenient *force scale* and draw *ab*, Fig. 78, to represent AB to this scale, *ab* being parallel to AB. From *b* draw *bc* parallel to BC and to scale. We can now close the *triangle* or *force polygon* by drawing *ca*. It will be found, by measurement, that *ca* represents CA to the force scale employed and, further, that *ca* is parallel to the line of action of CA.

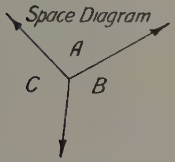


Fig. 77

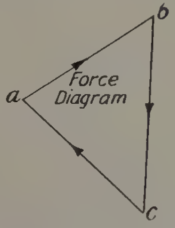


Fig. 78

The experimental proof of this theorem is very useful, and forms part of the laboratory training of an engineer.

Observe that in the space diagram, Fig. 77, we use capital letters, placing one in *each space*; a force is referred to by the two letters, one on each side of the line which represents the force. In the force diagram, Fig. 78, we use small letters and place these at the *ends* of the lines. This method, known as Bow's Method of Notation, is a convenient one, and will be used throughout these lessons where required.

Polygon of Forces

An extension in the number of forces considered will lead to a polygon, every side of which represents completely the corresponding force. For example, suppose the forces AB, BC, CD, DE, EF, and FA, acting as shown in Fig. 79, to be in equilibrium, then the polygon *abcdef*, Fig. 80, represents these six forces. The point of application of each of these forces is known; there remain the further three quantities with which we are concerned, viz., magnitude, direction, and sense. The direction of the force is given by the slope of the line representing it, and the sense by the arrow head. The number of unknown quantities which it is possible to determine is limited, and, since in practical problems we are usually concerned with the magnitudes of the forces, we will assume that we know the magnitude and direction of AB, BC, CD, and DE, and require the magnitude of EF and FA, their directions being known. The method employed is identical with that used when dealing with the triangle of forces.

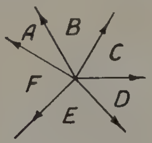


Fig. 79

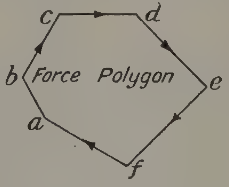


Fig. 80

Selecting a suitable force scale, we set down *ab*, Fig. 80, to represent AB, *bc* to represent

BC, *cd* to represent *CD*, and *de* to represent *DE*. To determine the forces *EF* and *FA* we must close the force polygon. From *e*, draw a line parallel to the known direction of *EF*, and from *a* draw a line parallel to the known direction of *FA*. These lines cut at *f* and so close the polygon. *ef* and *fa* measured to the force scale give the magnitudes of the forces *EF* and *FA* respectively.

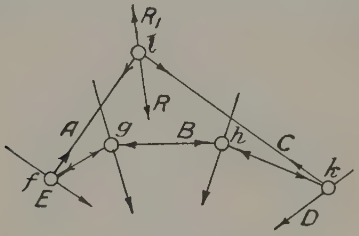


Fig. 80 (*a*)

Funicular Polygon

Reciprocal Figures (Fig. 80 (*a*)). Let *EA*, *AB*, *BC*, and *CD* be four forces which act upon a body. Further, let the body be acted upon by a single force *DE*, which, together with the four forces mentioned above, produce equilibrium. The magnitude of this single force *DE* must equal the *vector sum* of *EA*, *AB*, *BC*, and *CD*, since it produces equilibrium. To obtain this *vector sum* we add four forces *ea*, *ab*, *bc*, and *cd* as in Fig. 80 (*b*); the *vector sum or resultant* is *ed*, and hence, the *equilibrant*, or that force which, together with *ea*, *ab*, *bc*, and *cd* produces equilibrium, is *de*.

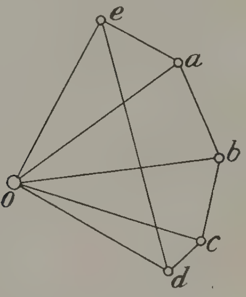


Fig. 80 (*b*)

We have now to determine a point on the line of action of this resultant. Select any pole point o; join *oe*, *oa*, *ob*, *oc*, and *od*. From any point *f* on the line of action of *EA*, draw *fg* parallel to *oa*; *gh* parallel to *ob*, and *hk* parallel to *oc*. From *k* draw *kl* parallel to *od*, and from *f* draw *fl* parallel to *oe*. Then *l* is a point on the line of action of the resultant $ed = R$. We may regard the figure *lkhgf* as composed of five metal or other links in equilibrium under the action of the applied loads *EA*, *AB*, *BC*, and *CD*, and the equilibrant R_1. The applied loads will set the links *fg*, *gh*, and *hk* in compression, and these links are kept in equilibrium by the two links *fl* and *kl* which are thus put in tension. Our system at *l* is now reduced to the tensions in the two links *lk* and *lf* and the equilibrant R_1. Since these are in equilibrium they must meet in a point, that is, the resultant or equilibrium passes through *l*.

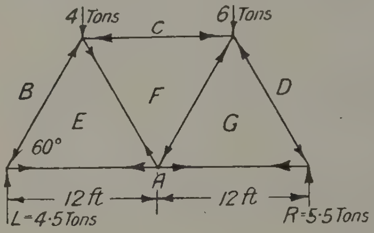


Fig. 81

The figure *fghkl* is termed the *Funicular* or *Link Polygon* for the forces *EA*, *AB*, *BC*, *CD*, and *DE*. Also Fig. 80 (*a*) is *reciprocal* to Fig. 80 (*b*), since for every line in one figure there is a corresponding parallel line in the other.

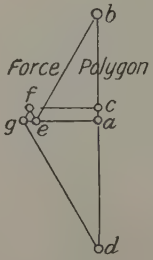


Fig. 82

Graphical methods find their chief application in the solution of *articulated* structures, such as roof frames, lattice girders, etc., and in order to make the subject clear we shall give the solution of a number of examples.

Example. A Warren girder of two bays, each 12 ft. long, carries two loads as shown in Fig. 81. The girder is supported at its ends; determine graphically the force acting on each member of the girder.

Method. The reactions can be obtained graphically by considering the girder as an ordinary beam and using the method explained in Lesson VIII, drawing the line of loads and the funicular polygon; or, we can obtain the reactions by a simple calculation of moments—

$$R \times 24 = (4 \times 6) + (6 \times 18)$$
$$\therefore \quad R = 5{\cdot}5 \text{ tons and } L = 4{\cdot}5 \text{ tons.}$$

We must now *letter* diagram Fig. 81 by placing a letter—A, B, C, D, E, F, and G—between each force and in each space as shown. Selecting a suitable load scale, set down the line of loads *bcd*, and mark on this line the point *a*, making $da = DA = R = 5{\cdot}5$ tons. In drawing the force polygon, Fig. 82, the reader must confine his attention to one joint at a time, and to those forces which act at that joint. Confusion is sometimes caused by trying to include bars which are adjacent, but are not connected, to the joint under consideration. It is a good method, when working this type of problem, to *use both hands*, the right-hand for the force diagram, Fig. 82, and the left-hand for the space diagram, Fig. 81. We must decide upon the order in which we intend to travel around the joints; either clockwise or anticlockwise will serve, but, having once started, we must retain that direction throughout the problem. We will select the clockwise direction, so that, taking the joint at the left-hand support, the forces will be AB, BE, and EA, not BA, AE, and EB, as would be the readings for anticlockwise direction. From *b*, Fig. 82, draw *be* parallel to BE, Fig. 81, and from *a*, draw *ae* parallel to AE. *abe* is a triangle; the sides of which, when measured to scale, give the magnitudes of the forces acting on the members BE and EA. We have now to determine the directions of these forces. Using the right hand for Fig. 82, we commence with the known reaction *ab*, in so doing the *right* hand moves *upwards* and this is the direction of the known reaction. From *b* to *e*, the right hand moves *downwards towards the left*, and in this direction we place the arrow head on the bar BE; again, from *e* to *a* the right hand moves from *left* to *right*, and this gives the direction of the arrow head which we now place on the bar EA. We can now place an arrow head on the *other* ends of the bars BE and EA; these arrow heads indicate the directions of the forces exerted by the bar in each case. Thus, on bar BE, the arrows point *outwards*, indicating that the bar is acted upon by two external compressive forces and the reaction of the bar causes it to thrust *outwards* as shown by the arrow heads; clearly then, BE is a strut subject to end compressive forces. On the bar EA, the arrow heads point inwards, giving the direction of the forces which the bar exerts when reacting to the externally applied forces which set the bar in tension.

Having determined the force in BE, the number of unknown forces at the joint carrying the 4 ton load is reduced to two, and so this joint is now ripe for graphical solution. In Fig. 82, from *c*, draw *cf* parallel to CF, and from *e*, draw *ef* parallel to EF. *cf* and *fe*, when measured to the force scale, give the forces in the bars CF and FE respectively. Again going around the joint in Fig. 81, with the *left* hand in clockwise direction and using the *right* hand for Fig. 82, we find that from *c* to *f* the *right* hand moves from right to left, and we place the arrow head on CF in this direction. Similarly, from *f* to *e*, the *right* hand moves *downwards towards the right*, and this enables us to place the arrow on the bar EF as shown. Again the arrow can be placed on the other ends of the bars CF and FE; it is now clear that CF is a strut, and EF is a tie. Proceeding, as already explained, we deal with the three remaining joints and obtain the complete force diagram, Fig. 82. Scaling off the lines in Fig. 82 and noting the directions of the arrow heads in Fig. 81, we are able to tabulate the forces on the bars and the kind of force, tensile or compressive, as follows—

Bar	Force in Tons
BE	+ 5·19
EA	− 2·596
CF	+ 2·88
EF	− 0·577
DG	+ 6·35
GF	+ 0·577
AG	− 3·176

+ indicates a strut.
− indicates a tie.

The following example of a Warren girder is given as a problem which the reader is urged to solve for himself, the answers, written on the bars themselves, being given for reference only.

EXAMPLE. A Warren girder of three bays is loaded as shown in sketch, Fig. 83. Determine, graphically, the load in each member of the frame and tabulate your answers, indicating struts and ties.

ROOF TRUSS

The graphical solution of a loaded roof truss can now be undertaken; it will be found to be a simple extension of the previous examples, involving, as it does, the *taking of moments* about the ridge or apex, in order to determine the force in the horizontal tie at mid span.

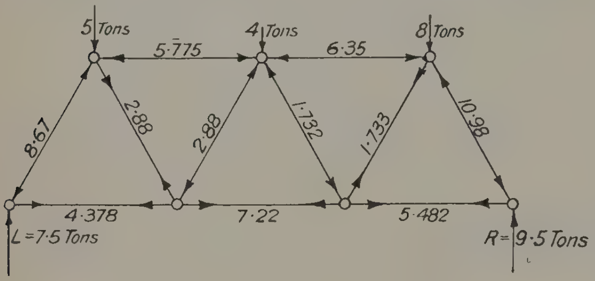


FIG. 83

EXAMPLE. A roof truss is pitched at 40° and spans 40 ft. The arrangement of the bars and the loads on the truss are shown in Fig. 84. Draw the complete force polygon or diagram, and tabulate the bars, and the loads in the bars, indicating struts and ties.

REACTIONS. These are obtained by the simple calculation of moments of forces about either support, say, about *L*—

$$R \times 40 = (2 \times 5) + (4 \times 10) + 5 \times 15) + (6 \times 20) + (6 \times 25) + (6 \times 30) + (4 \times 35)$$

$$\therefore \ R = 17{\cdot}875 \text{ tons and } L = 15{\cdot}125 \text{ tons.}$$

Method. Set down the line of loads *abcdefgh* to any convenient load scale. Fix the point *k* by making *hk* = *R* = 17·875 tons to scale. From *a*, draw *am* parallel to *AM*; from *k* draw *km* parallel to *MK*; the intersection gives the point *m*. Using the right hand for Fig. 85 and the left hand for Fig. 84, as described previously, we see that *AM* is a strut and *MK* is a tie. Taking next the joint at which the 2 ton load acts, we proceed to determine the forces in *BN* and *NM*. From *b* draw *bn* parallel to *BN*; from *m* draw *mn* parallel to *NM*; the intersection gives the point *n*. *BN* is seen to be a strut and *NM* is also a strut. We can now take the lower joint and determine the forces in *NO* and *OK*, having already determined *KM* and *MN*. From *n* draw *no* parallel to *NO*, and from *k* draw *ko* parallel to *OK* (this line has already been drawn); the intersection gives the point o. Both *NO* and *OK* are seen to be ties. We have now come to a temporary halt in our graphical work; if we seek to solve the joint at which the 4 ton load *BC* acts, we find that there are *three* unknown forces, viz., *CQ*, *QP*, and *PO*, and our graphical method is inapplicable. Again, the joint formed by the junction of the bars *KO*, *OP*, *PT*, and *TK*, presents the same difficulty, there are *three* unknown forces, viz., *OP*, *PT*, and *TK*. An easy method of surmounting this difficulty is by *taking moments* of all the external forces on

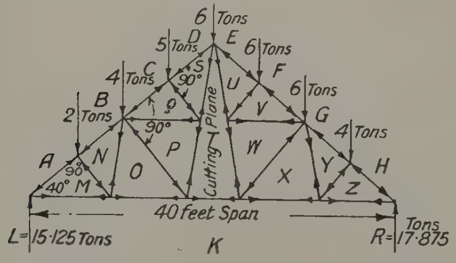


FIG. 84

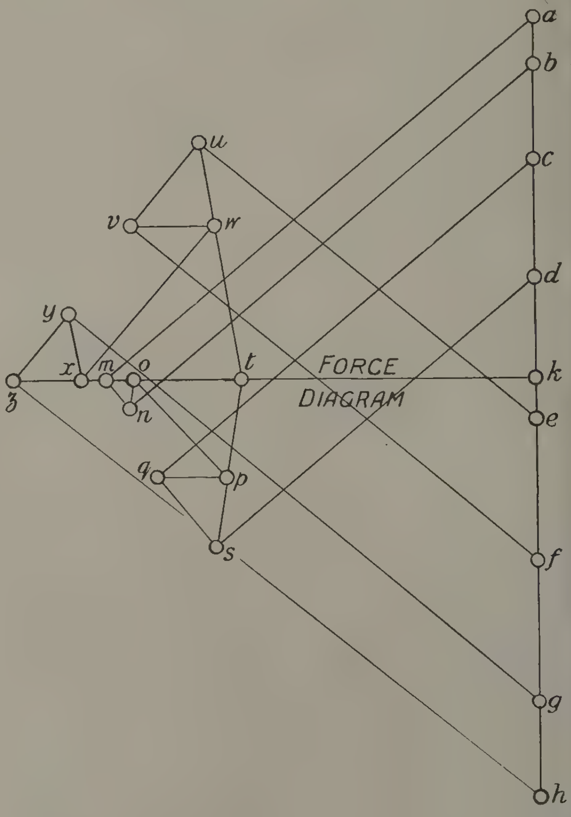


FIG. 85

either half of the truss, which we can assume to be cut by a vertical plane, which passes through the apex, and severs the horizontal tie bar TK. Let the reader cover over, say, the right half of Fig. 84 with a piece of blank paper, the edge of the paper passing through the top or apex and cutting the horizontal tie; this is shown in Fig. 84 by the line marked *cutting plane*. Now consider the equilibrium of that part of the truss exposed to view. It is acted upon by the three applied loads, AB, BC, and CD, each having an anticlockwise moment about the apex joint; the reaction $L = 15{\cdot}125$ tons, which exercises a clockwise moment about the apex; and, further, there is the force in the severed tie bar TK. We can now take moments as follows—

$$15{\cdot}125 \times 20 = (2 \times 15) + (4 \times 10) + (5 \times 5) + (TK \times 16{\cdot}782).$$

$$\therefore\ TK = \underline{12{\cdot}37 \text{ tons.}}$$

Having calculated the force in the horizontal tie bar TK, we can mark the point t, on the horizontal line through k. We are now able to solve the joint KO, OP, PT, and TK, as there are now only two unknown forces, viz., the forces in OP and PT. From o, draw op parallel to OP; from t, draw pt parallel to PT. The intersection of these lines gives the point p, and we decide that OP is a strut and PT is a tie. The next joint to be solved is that at which the 4 ton load BC acts, and here we now have only two unknown forces, so that our graphical method is sufficient to provide the solution. The remainder of the problem presents no further difficulty and, provided care is exercised in drawing the necessary parallel lines, etc., the forces in the bars can be determined with great accuracy. The complete solution is tabulated as under—

Bar	Force in Tons	Bar	Force in Tons	Bar	Force in Tons
AM	+ 23·68	*EU*	+ 18·6	*KM*	− 18·13
BN	+ 22·4	*FV*	+ 22·44	*KO*	− 16·92
CQ	+ 21·26	*GY*	+ 25·82	*KT*	− 12·37
DS	+ 18·0	*HZ*	+ 28·0	*KX*	− 19·16
MN	+ 1·55	*PT*	− 4·5	*KZ*	− 22·04
NO	− 1·24	*ST*	− 7·55	*VW*	− 3·57
OP	+ 5·8	*TU*	− 10·33	*WX*	+ 8·72
PQ	− 2·97	*TW*	− 6·73	*XY*	− 2·88
QS	+ 3·85	*UV*	+ 4·6	*YZ*	+ 3·68

+ indicates a strut; − indicates a tie

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XV

Academical and Practical Drawing. We have, in a previous lesson, observed that economy of time can be effected in the making of a drawing by substituting approximate but quickly produced lines for more accurate but tediously produced lines and projection. We may add that progress in workshop practice tends to increase the amount of line in a drawing which may be approximate or entirely eliminated.

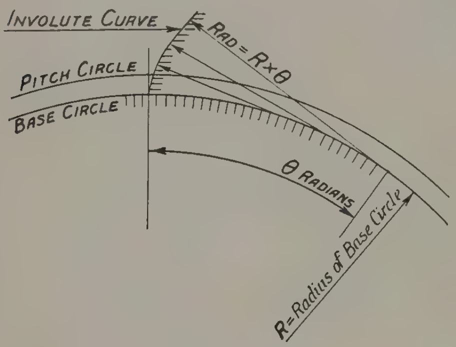


Fig. 1

Whilst, in the interest of economy, we ought always to take advantage of any opportunity to curtail the work on a drawing, we must not overlook the advantage the more precise drawing offers in helping one to visualize the actual form and to appreciate the building up of the correct construction.

Academical Drawing. Drawing, which is precise beyond the needs of the craftsman constructing the machine part, may be termed *academical*. As an example of such drawing we may take the case of a machine part containing spur gear teeth. Leaving for the present the question of how much information is required to allow the shops to proceed with the manufacture of the detail, we may consider the geometrical construction of a tooth form.

The Involute. The tooth form most commonly adopted for spur gearing is the involute shown in Fig. 1. The outline of this tooth form is produced by a point on a cord in the process of being unwound from a cylinder. The circle representing the cylinder is termed the *base circle*. Referring to Fig. 2, the two base circles shown produce the tooth form of two spur wheels in gear. The line *ab*, which is a common tangent to the two base circles, is termed the *line of pressure*, and the angle *aoc* is termed the *pressure angle*. The point *c*, where the line of pressure intersects the line joining the axes of the two wheels, is termed the *pitch point*, and the circle concentric with the base circle and passing through the pitch point is termed the *pitch circle*. It is an essential condition, to obtain uniform motion, that the common normal to the teeth in contact, at the point of contact, must pass through the pitch point. In the involute tooth form the line of contact coincides with the line of pressure, and so the above condition is fulfilled.

The height of the tooth above the pitch circle is called the *addendum*, and the depth of the tooth below the pitch circle is termed the *dedendum*. If an arc, Fig. 2, be drawn at a distance of the addendum above the pitch circle of the driven wheel, intersecting the line of pressure at *d*, then *dc* is termed the *arc of approach*. If, further, a corresponding arc be drawn at a distance of the addendum above the pitch line of the driver intersecting the line of pressure at *e*, then *ce* is termed the *arc of recess*. The line *de* is called the *arc of contact*.

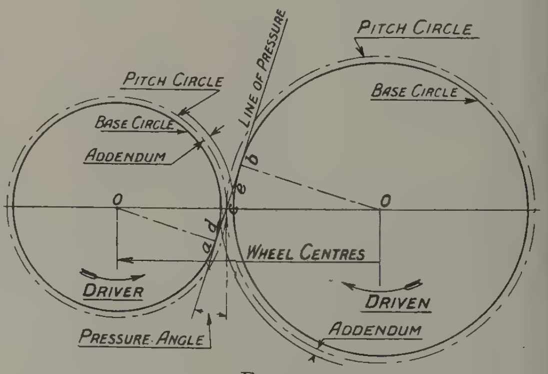


Fig. 2

The involute curve, of course, finished at the base circle and where, as in the general case, the dedendum lies below the base circle, this portion of the tooth form is formed by a tangent to the involute passing through the wheel axis.

If the addendum on the driven wheel is such that the intercept *dc* on the pressure line is greater than *ac*, interference would result in that portion of the tooth below the base circle, and to enable the teeth to mesh the lower part of the teeth would require to be undercut. Undercutting produces a weaker tooth and the loss of a part of the tooth profile, and consequently should, where possible, be avoided.

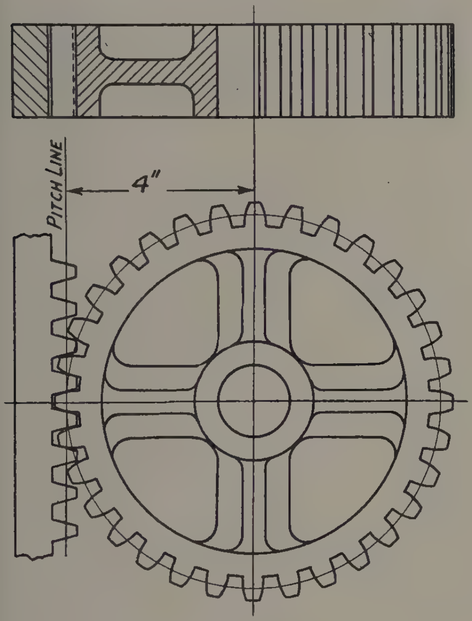


FIG. 3

Rack and Pinion. Suppose we have to make a large scale drawing of an involute tooth form for a pinion, to gear with a rack as Fig. 3.

A full size detail for a tooth in the above wheel would obviously be too small to allow an accurate outline of the tooth form to be drawn with any degree of accuracy. To obtain precision we must increase the scale above the normal. Making the scale, 10 in. = 1 in., we encounter at the outset a difficulty which, being common to many machine drawings, is worthy of our attention. Commencing the drawing with the pinion base circle, and adopting a pressure angle of $14\frac{1}{2}^\circ$, we have the base circle diameter

$$= PCD \times \cos 14\tfrac{1}{2}^\circ = 8 \text{ in.} \times {\cdot}9681$$
$$= 7{\cdot}7448 \text{ in.}$$

The actual length of line required to represent the radius of the base circle

$$= \frac{7{\cdot}7448}{2} \times 10 = 38{\cdot}724 \text{ in.}$$

The limiting radius for most pencil compasses is 8 in., and for the average set of trammels 24 in. We may, therefore, consider the alternative method of producing an arc from plotted points. Referring to Fig. 4, the arc *cp* has a common radius *oc*. A point *p* on the curve may be given relative to the point *c* in terms of the rectangular co-ordinates *x* and *y*. When the angle subtended by the arc *cp* is θ°, then

$$y = oc \times \sin \theta^\circ.$$
$$x = oc - oc \times \cos \theta = oc\,(1 - \cos \theta).$$

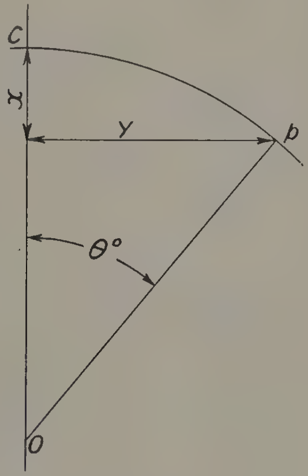


FIG. 4

Tabulating values of the above ordinates at various angles for an arc of 3·8724 in. radius, as in Table I, we may commence the drawing with a portion of the pinion base circle as in Fig. 5. The arc between each of the above plotted points subtends an angle of 4°, and, commencing with the point *c*, where the involute radius is zero, the involute radius from each point to the right of *c* increases by an amount equal to

$$\frac{4^\circ}{360^\circ} \times 2\pi R = \frac{3{\cdot}8724\pi}{45} = {\cdot}2704 \text{ in.}$$

We may now draw lines tangential to the base circle at the plotted points and, setting off from the same points the corresponding involute radii, obtain a number of points on the involute curve. Drawing the line *cv* through the points we have the involute for the given base circle.

The pitch circle may now be drawn at a distance of $4 \text{ in.} - 3{\cdot}8724 \text{ in.} = {\cdot}1276 \text{ in.}$ above the base circle, cutting the centre line *cd* at *e*, and a line *ef* drawn normal to the line *cd* will represent the pitch line of the rack.

We have now to determine a suitable value for the circular pitch of the teeth. It is evident that the two variables, circular pitch and addendum, may, within limits, have various values for the same wheel. The maximum addendum of the rack, to avoid undercutting, is fixed by the perpendicular distance of the point *j* from the rack pitch line. Having found the maximum addendum, the corresponding minimum pitch would be determined by the point of intersection of the two involute curves *cv* and *gh*. For convenience in manufacture a standard proportion of addendum to pitch has been adopted for machine cut gears. Using Messrs. Brown & Sharpe's standard proportion we may find the maximum circular pitch giving the strongest tooth.

TABLE I

REFER TO FIG. 4

Angle $\theta°$	4	8	12	$14\frac{1}{2}$	16	20	24
Cos θ	·9976	·9903	·9781	·9681	·9613	·9397	·9135
1–cos θ	·0024	·0097	·0219	·0319	·0387	·0603	·0865
$3{\cdot}8724\,(1-\cos\theta) = r$ in.	·0093	·0376	·0848	·1235	·1498	·2335	·3350
Sin θ	·0698	·1392	·2079	·2504	·2756	·3420	·4067
$3{\cdot}8724 \sin\theta = y$ in.	·2704	·5391	·8052	·9692	1·0670	1·3240	1·5750
Radius R in.	·2704	·5408	·8112	—	1·0816	1·3520	1·6224

$$\text{Addendum} = {\cdot}3183 \text{ in.} \times \text{circular pitch}$$

$$\therefore \text{ max. pitch} = \frac{{\cdot}2511}{{\cdot}3183} = {\cdot}7891$$

The corresponding number of teeth in the pinion would be

$$= \frac{\pi \times PCD}{\text{circ. pitch}} = \frac{8\pi}{{\cdot}7891} = 31{\cdot}85$$

Deciding upon 32 teeth we find the actual

$$\text{circular pitch} \quad = \frac{8\pi}{32} = {\cdot}7854 \text{ in.}$$

$$\text{and the actual addendum} = {\cdot}7854 \times {\cdot}3183 = {\cdot}25 \text{ in.}$$

$$\text{Also the dedendum would} = {\cdot}7854 \times {\cdot}3683 = {\cdot}289 \text{ in.}$$

We may now mark off the thickness of the pinion tooth *kl*, which measured along the pitch circle will be equal to one-half the circular pitch. Having drawn a line at a distance of the actual tooth addendum above the pitch circle we may now trace the involute curve through the point *e*. Further, drawing a line at a distance of the actual dedendum below the pitch circle, we may complete the pinion tooth profile. Proceeding to the rack we observe the pitch line to be straight, and consequently the curvature of the involute will be zero, i.e. the involute will be a straight line. We may now proceed to outline the rack tooth form, and so complete the drawing with the tooth forms shown shaded, as in Fig. 5.

EXERCISE

Make a large scale drawing of the tooth form for the above rack and pinion, allowing a pressure angle of 16°.

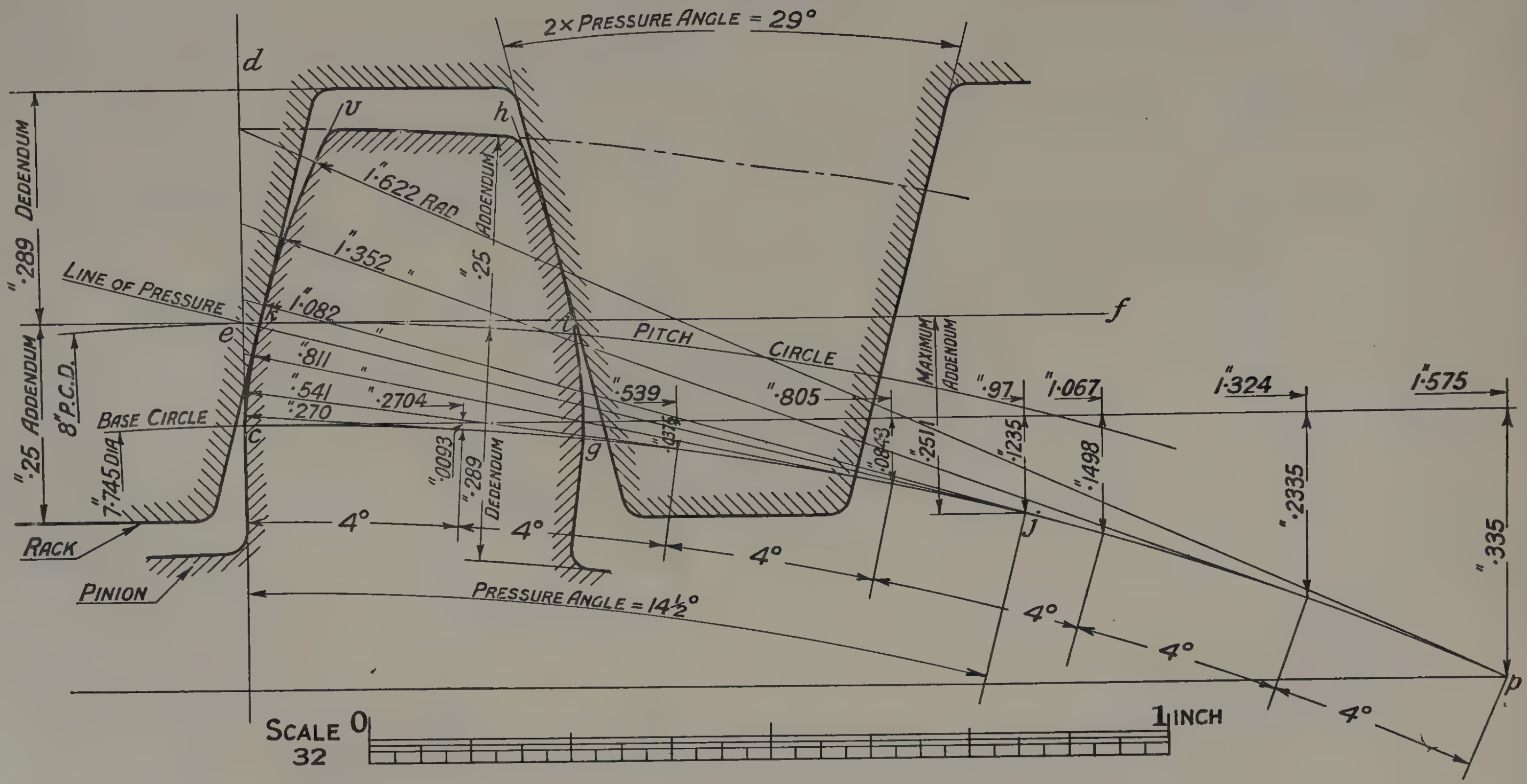


Fig. 5. Large Scale Construction of Involute Tooth Form

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.I.N.A.

LESSON V

EXAMPLES OF TURBINE CONSTRUCTION

The de Laval Turbine

The best known example of the smaller powered turbines is probably the de Laval. This is very fitting, because its inventor, Dr. Gustav de Laval, showed some genius in the masterly solution of problems that arose during the development of a very high speed turbine.

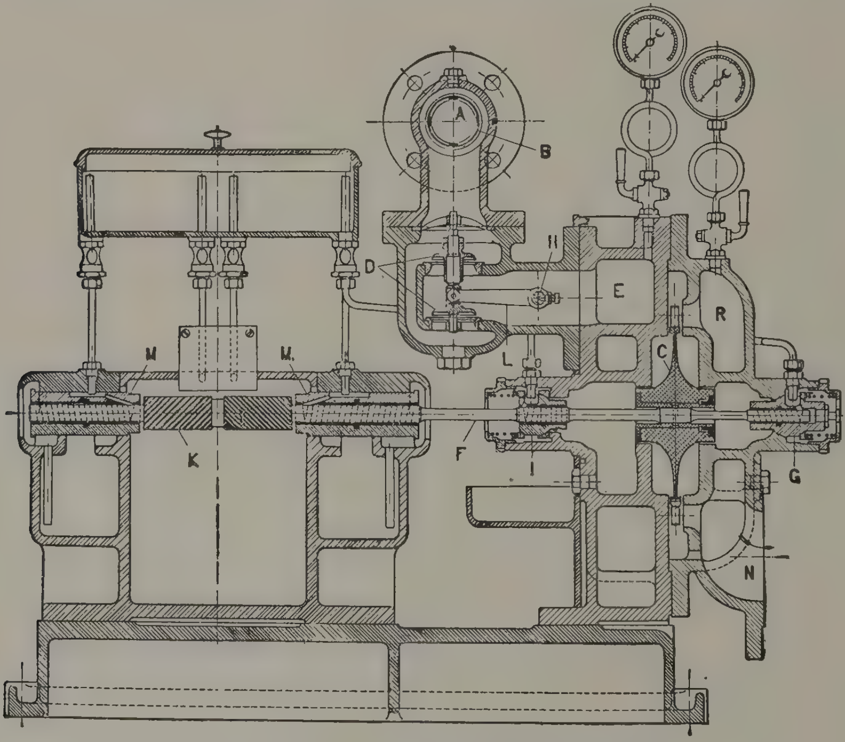


Fig. 17. 30 H.P. de Laval Turbine
Sectional view

We have already seen that the efficiency of a steam turbine of the impulse type depends mainly upon the ratio which the peripheral velocity of the blades bears to the velocity of the steam at exit from the nozzle. Let us consider a simple impulse turbine, such as the de Laval, supplied with live steam at 100 lb. per sq. in. gauge, superheated 50° F. Further assume that the vacuum in the exhaust branch is 25 in. of mercury with a 30 in. barometer, i.e. that the exhaust pressure is 2·45 lb. per sq. in. absolute. Then, by means of the Mollier chart, we find that the "heat drop" or energy theoretically available is 263 B.Th.U.'s per lb. of steam. About 13 per cent of this energy would be lost in friction and other losses in the nozzle giving 229 B.Th.U.'s as the effective gain of kinetic energy per lb. of steam.

$$\therefore \text{ exit velocity } = \sqrt{2g \times 778 \times 229}$$
$$= 3390 \text{ ft. per second.}$$

The angle made by the nozzle axis with the plane of the wheel in the de Laval turbine is sually 20°, so that the theoretically most fficient value of ρ is 0·47.

Therefore, most efficient blade velocity

$$= 0{\cdot}47 \times 3390$$
$$= 1590 \text{ ft. per second,}$$

r over a thousand miles an hour. This is an xceptionally high speed for turbine rotors, lthough speeds approaching this value have ctually been adopted in the higher powered e Laval turbines. It is not economical to un smaller machines, say, up to 50 h.p. at this peed owing to the increased first cost, and peeds of 600 to 800 ft. per second are more eneral.

Even at these peripheral velocities, unless the urbine rotor is made large in diameter, the otational speed is very high. Again, considerations of first cost, particularly of the forged teel rotor, demand that the diameter shall be s small as possible, and, as a result, the de Laval urbine is invariably a high speed machine. The otational speed varies from about 10,000 r.p.m. n the older 300 h.p. machines (now obsolete) up to 40,000 in the smallest turbine generating $1\frac{1}{2}$ h.p. For this reason, even though the turbine is used for driving generators, fans, and pumps, it is invariably geared down through double helical reduction gearing.

Figs. 17, 18, and 19 show a 30 h.p. de Laval turbine.

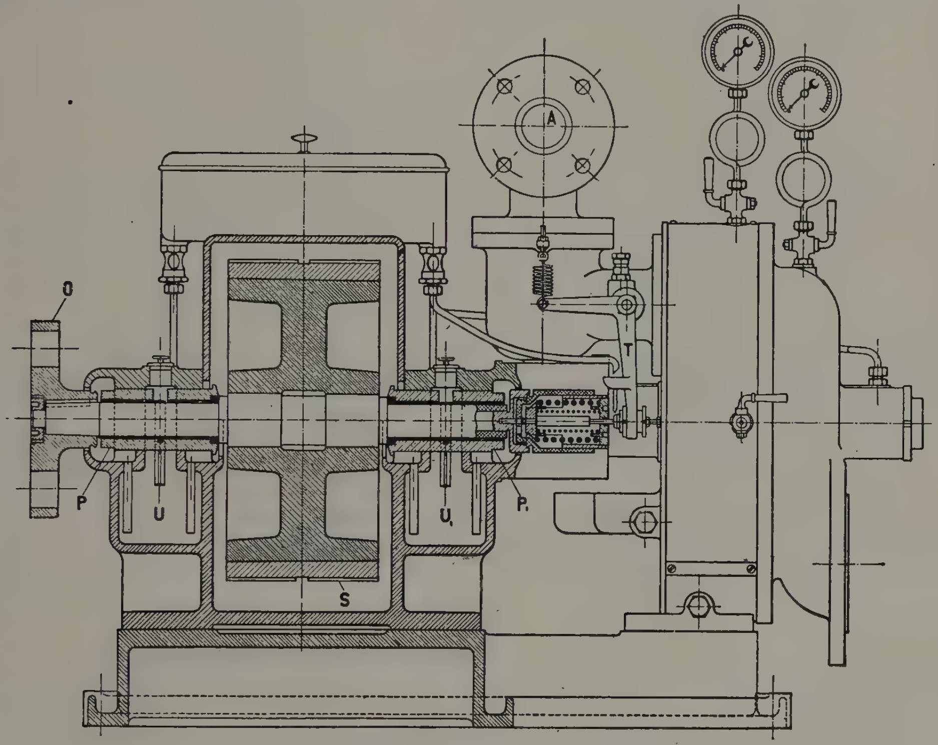


FIG. 18. 30 H.P. DE LAVAL TURBINE
Section through main gear wheel and governor

Live steam supplied through the stop valve to the branch *A* passes through the strainer *B*, which prevents scale and other foreign matter from choking up the nozzles or blades. It then flows through the balanced governor valve *D* into the steam chest *E*, from whence it flows via the nozzles into the blade channels, and finally into the exhaust belt *R* leading into the exhaust branch *N*.

The inlet to each nozzle is controlled by a mitre-seated valve, as shown in Fig. 20. There are usually two sets of nozzles, one to be used when the turbine is operating on atmospheric exhaust, and the other to be used when the turbine is exhausting into a condenser. The dimensions of the nozzles are such that full power may be developed on either group of nozzles alone. The nozzles are of the convergent-divergent type, with the throat quite near the inlet and having a gradual taper in the diverging portion.

The method of attaching the blades to the rotor will be followed from Fig. 21. The rotor rim has first a number of holes *A* drilled parallel to the axis. The slot for the blade root is then finished by milling. The blades are drop-forged steel and their roots are afterwards finished by milling. Each blade is put in separately and may, if it should prove necessary, be replaced without disturbing any other blades. The working section is, of course, curved and the root is rectangular, the transition from one section to the other being made by the "landing" *B*. Each blade is provided with a "roof" or "shroud" *C*. These are fitted very carefully so that when all the blades are in position a continuous ring of shrouding is formed, which tends to prevent both blade vibration and radial spilling of the steam.

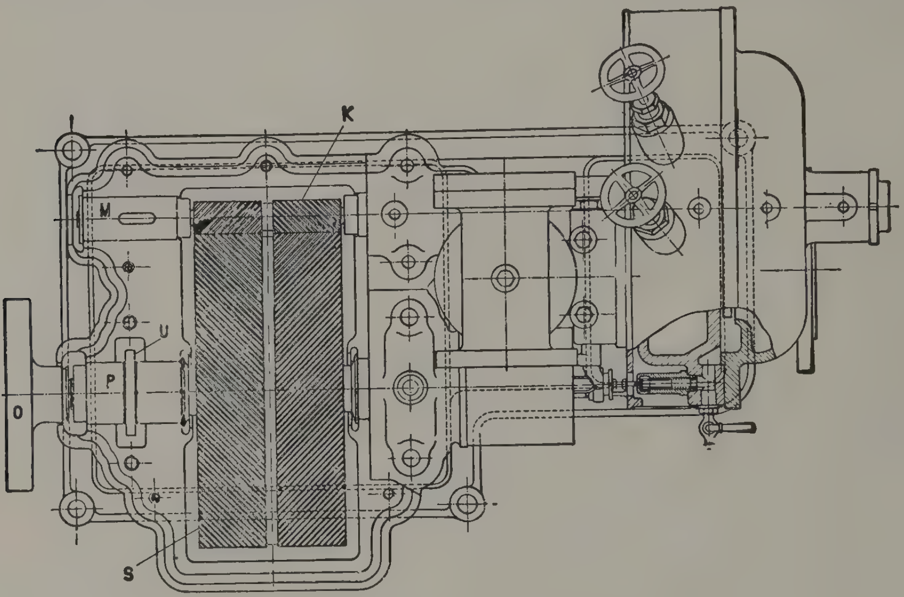


Fig. 19. 30 H.P. de Laval Turbine
Plan view, with gear case cover removed

For moderate speeds, the rotor disc is bored to fit the spindle as shown in Fig. 17. For the higher speeds the disc is made solid and, in consequence, is very much stronger. In this case, the spindle must be made in two halves, each half being attached to a boss on the side of the disc.

The long slender spindle on which the rotor is mounted is one of the remarkable features of this turbine. Restrictions of space forbid anything but the briefest mention of it, but it may be said that when a shaft or spindle supports a single wheel, then as its speed increases from rest, unless the rotating system is perfectly balanced (and it never is in practice, despite the most refined methods), it will commence to vibrate with a whirling motion when it approaches a certain rotational speed, known as the *critical speed* or *whirling speed*. There is a definite whirling speed for every turbine rotor, depending on the disposition and weight of the various rotating masses, and on the elastic properties of the supporting spindle. It may be mentioned here that the designer must always determine, by calculation, what this speed is, and make quite certain that it is well removed from the running speed of the machine. In the general case, there are several critical speeds, viz., one fundamental and the harmonics, but in the case of a light spindle supporting a single disc which is relatively heavy there is only one critical speed and, once this speed is exceeded, the rotor spins in a stable condition with a stability which increases as the speed increases. In de Laval turbines, the whirling speed is generally about one-seventh of the normal running speed. Thus, the whirling speed is very quickly passed during the starting up of the turbine, and no damage is done due to the inevitable vibration.

In the turbine illustrated in Fig. 17, the rotor spindle *F* is extended to form the pinion *K* of the reduction gear. The spindle is supported in long brass bushes lined with white metal. In the example shown, lubrication is effected by sight-feed syphon lubricators, but in the larger machines forced lubrication is now adopted. The oil is distributed along the bearings in the example shown by means of a spiral groove cut in the white metal and effected by the viscous drag of the shaft surface. In the special bush *I*, which also functions as a gland, only a few spirals are provided. In addition there are three or four plain grooves which assist in preventing air leakage into the turbine casing.

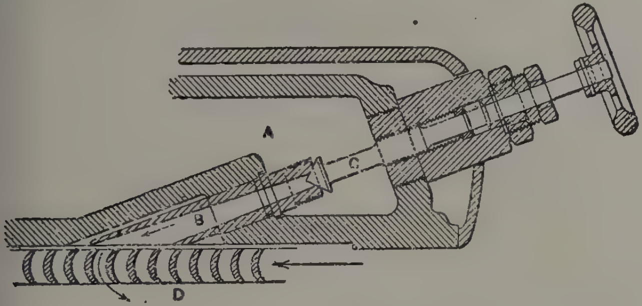


Fig. 20. De Laval Turbine Nozzle

The main gear wheel, Fig. 18, has a cast-iron centre and a steel rim shrunk on.

The method of regulating the lift of the throttle valve *D*, Fig. 17, so as to maintain a fairly constant speed will be apparent from Figs. 17 and 18. A centrifugal governor of ingenious design, loaded by a central spring, is attached to the low speed shaft. A small central spindle, corresponding to the sleeve of an ordinary governor, runs in contact with the lower end of the bell crank lever *T*, Fig. 18. The lever *T* is pinned to the spindle *H*, Fig. 17, and therefore operates, by its movement, the throttle valve *D*. If an increase in speed occurs, due to a reduction of load, the half-cylindrical weights of the governor move to a slightly larger radius and, through the medium of the small central spindle, turn the lever *T* in the counter-clockwise direction, thus reducing the lift of the throttle valve *D* in accordance with the reduction of power.

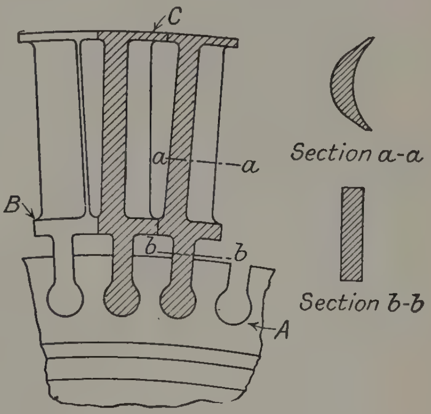


Fig. 21

A further emergency governor consists of a butterfly valve in the exhaust pipe, but this cannot be described here.

The efficiency ratio of the de Laval turbine varies from about 35 per cent in the small sizes up to about 60 per cent in the largest sizes, the figures referring to simple turbines.

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON I

INTRODUCTORY REMARKS

At the present day no engineer worthy of the name can afford to be content with less than a good working knowledge of the testing of prime movers. The competition between the various types of prime movers, and between the different manufacturers of such plant, is now so keen that in nearly all important installations guarantees are required, or are given, respecting the performance, efficiency, and reliability of the plant. In many cases penalties are provided if the guarantees are not fulfilled under test, with a corresponding proviso that a bonus shall be paid to the maker if the guarantees are more than fulfilled. The plant may be tested in the maker's works before delivery, or after being installed on the site following on some specified period of operation under running conditions, or again, in some cases, both sets of tests may be specified. It is further the ordinary practice in important power stations to measure continuously the principal items of input and output, so as to be able to ascertain the performance and efficiency of the station, in whole, or in part, under ordinary working conditions. Not only is it important, therefore, to be able to test power plant effectively, but it is equally important to be able to interpret the results of such tests, whether the tests are made personally or by others.

Most of the tests on power plant are concerned largely with the thermal and mechanical efficiencies, or with the corresponding steam and fuel consumptions, at different loads. It is not always realized how important may be the advantage to be derived from even a relatively small increase of thermal or mechanical efficiency. In a power station, for example, a 5 per cent increase in the thermal efficiency of the steam turbines, resulting in a corresponding reduction in the consumption of fuel, not only means an increased economy of fuel, but usually means a 5 per cent reduction in the number and cost of the boilers. Again, an improvement in the thermal efficiency of an oil engine usually results in a corresponding reduction in the amount of heat rejected by the engine, and makes for greater reliability by reducing the chance of pistons and valves, etc., becoming overheated. The highest possible mechanical efficiency of each part of a power plant should also be desired consistent with reliability. If an engine were driving a dynamo, for example, to supply electrical power to motors, an individual 5 per cent increase in the mechanical efficiency of each part of the plant would mean a probable increase in the overall efficiency of something like 15 per cent, because the overall efficiency is the product of the several efficiencies in the series.

When tests are to be made for thermal and mechanical efficiencies it is important to consider, beforehand, certain fundamental conditions, such as, the period of running required before the test commences, the duration of the test, and the observations to be recorded. The duration of the test should depend primarily upon the accuracy with which the principal items can be measured, upon the degree of steadiness obtainable during the test, and upon the degree of accuracy required in the result. The careful consideration of what readings should be taken during a test is necessary to get the best results. In a boiler test, for instance, it is not sufficient to measure only the fuel and water consumptions, and their corresponding heat values, temperatures, etc., without regard to the measurable losses of heat. However carefully such a test has been conducted the result is always open to suspicion, especially if it is found that the heat quantities are not capable of being reconciled. A not uncommon experience in boiler tests is to find particulars given which enable one to estimate the principal losses of heat, and to find that the heat given to the water plus the losses comes out appreciably greater than the estimated heat supplied by the fuel. The whole test is then liable to be discredited, especially when the experimenters have given the particulars of the test without themselves having tested the heat balance.

In making tests on prime movers it is usually necessary to take a series of readings at definite intervals of time, because of the fluctuation of conditions which nearly always occurs during a test. The intervals between readings may be five minutes for tests of one or two hours' duration, or ten or fifteen minutes if the test lasts several hours. The calculations are usually based upon the averages of these readings. It is important that all readings should be recorded systematically so that no dubiety exists as to the meaning or interpretation of any observation. In an important test the readings should be checked as the test proceeds by plotting the principal readings on a time base. Any departure of the plotted points from straight lines would indicate changing conditions during the test, or that errors of observation or calculation have been made. If the variations are due to changing conditions it is usually found that the primary cause of the change is reflected in other changes which it automatically induces in some or all of the other items of observation.

In some cases it may not be correct to base calculations on the average readings unless the readings are very steady. When a fair degree of steadiness in the conditions cannot be maintained, the person in charge should seek to arrange the conditions at the end of the test to be approximately the same as at the start, for when there is a difference between the finishing and starting conditions a considerable error might result unless the changes were allowed for.

To consider a simple illustration, suppose a water heater contains 1,000 lb. of water and that the flow is at the rate of 10 lb. of water per minute. Let the starting temperatures be 60° F. and 120° F. at inlet and outlet respectively, with the finishing temperatures 65° F. and 125° F., and average values at inlet 62·5° F. and at outlet 122·5° F. Apparently, the heat transmitted to the water per minute is $10 \times (122{\cdot}5 - 62{\cdot}5) = 600$ B.Th.U.'s. But the body of water, 1,000 lb., has also been raised in temperature by 5° F. between the start and finish of the test. If the test were of 30 minutes duration, and the heat required to raise the temperature of the metal were neglected, then the transmission of heat would be $600 + \frac{1000 \times 5}{30} = 600 + 167 = 767$ B.Th.U.'s per minute. For such conditions probably it would be advisable to make the duration greater than 30 minutes if a more reliable result were desired.

An intimate knowledge of the laws connecting the principal data in a series of tests is often useful in checking the degree of reliability of the tests. For example, there are definite laws connecting the steam consumption, steam pressure, and load in steam turbines. It would not be sufficient merely to tabulate the results of a series of tests at different loads on such plant without plotting certain of these results against one another. In a throttle governed steam turbine the steam consumption should give a straight line when plotted against the load or against the steam pressure, all other conditions being nearly constant, and any departure from this relation would indicate the unreliability of one or more of the tests. On the supposition that the points lie near to a straight line and that a sufficient number of points are available, a line placed in an average position among the points should be taken to represent, within the range of the tests, the true relations between the quantities plotted. Any further lines or curves connecting these plotted quantities should be derived from the principal line or curve and not from the plotted values themselves.

In important tests an official report usually has to be made, and should include a description of the plant tested together with the principal apparatus used, and also of the testing arrangements or connections. A line diagram of the testing arrangements, not necessarily drawn to scale, will usually save lengthy descriptions, because the scheme of connections can then be seen at a glance. The important readings taken throughout the test might be given plotted on a time base, which is much more preferable than a tabulation because the graphs readily show the degree of steadiness and consistency of the conditions. The average readings would be tabulated together with the results calculated from them, having regard to the principal objects of the test or tests. A summary of the principal deductions to be obtained from the tests is very helpful to those interested who may not have the time available to make a detailed study of the results.

An important qualification for those in charge of a test is to know the types of measuring apparatus best suited for the particular tests to be made, while giving due consideration to questions of cost and reliability.

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON VIII

CORE PRINTS

The importance of correct core prints cannot be exaggerated, and the correct printing of a pattern may very considerably affect moulding costs.

Top prints are a danger when closing the mould, but they are sometimes a convenience for venting. When the metal is thin and there is not a substantial bottom bearance, top prints steady the core. Top prints are not a disadvantage for repetition work when moulding is done on machines. It is a good practice to give top prints about $\frac{3}{16}$ in. taper to the inch of depth and shape of the core-box, so that the core will be an accurate fit.

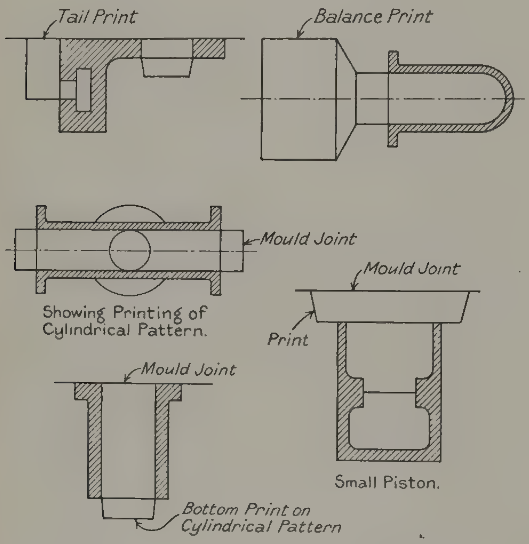

Tail Print
Balance Print
Mould Joint
Showing Printing of Cylindrical Pattern.
Mould Joint
Print
Mould Joint
Small Piston.
Bottom Print on Cylindrical Pattern

Fig. 25. Examples of Core Prints

Circular prints are the more common form of prints. When the pattern is jointed horizontally circular prints should be made in halves. Their thickness depends largely on the diameter of the print, and also on the shape and size of the core. A print may be a bearance, or it may merely serve as a guide. If as the latter, $\frac{7}{8}$ in. to 1 in. is sufficiently thick as a general rule. Core prints on top or bottom faces of patterns should always be tapered.

It is sometimes impossible to support cores with prints at both ends, and it is often desirable to make a heavy balance print, so that the core will not require other support and ensure that it will not move when casting. It is also frequently necessary, if the shape of a core is irregular, to shape the print, and thus insure against any possibility of wrong setting of the core.

Drop, Draw, or Tail Prints. Tail prints facilitate the setting of cores that would otherwise involve the making of difficult joints. The commonest type of tail print is for circular cores. There should always be a considerable taper on the sides of tail prints.

The thickness of tail prints depends to some extent at least on the thickness of metal to be cored, and also on the size of the work. A common use of tail prints is in coring holes in pipe flanges. Sometimes it happens that tail prints have to be made for two cores in the same vertical plane, but at different distances from the joint of the mould; the top print is simply secured to the face of the bottom print, when it is intended that the moulder should fill up the tail impressions, or, for better class work, the one print to the position of the bottom hole is all that is required, providing a core-box is supplied from which the full shape of the tail piece can be made, having the cores representing the holes projecting from it.

Sometimes slab cores are used by the moulder to support a body of sand, as when making moulding boxes. Supposing a flange is required to be cast at the bottom and about the outside of a box, and it is not possible to draw the pattern work into the mould for the sides, then the flange is fixed in a framed box. The box is generally made about 12 in. long, and as many cores are made in it as are necessary to go round the box. These cores are baked and then placed against the pattern sides with the flanges removed and on a level bearance; sand can then be rammed about them and the mould proceeded with. No prints are necessary in this case. In Fig. 25 several different kinds of prints are illustrated.

DIFFERENT TYPES OF PATTERNS

Whenever it is practicable, the pattern-maker hould define the moulder's joint. In what nay be termed regular shaped patterns this is simple matter, but in many other cases it is ot so. For repetition work, the first cost is of ess importance, and, with small work especi-lly, the moulders' joint can usually be defined vhen constructing the pattern. With some atterns it is possible to indicate the joint, and n other cases, such as with pipes and certain alves, while the body of the pattern may not e jointed, flanges and prints are jointed, a ractice which assists the moulder very naterially.

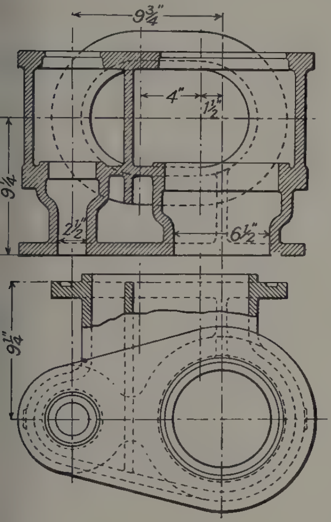


FIG. 26

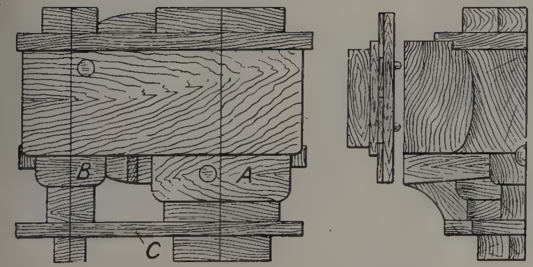


FIG. 27. A COMBINED VALVE CHEST

In many cases there is a choice between naking a solid or a skeleton pattern. The ecision is usually based on the quantity of astings required. For cylindrical work a attern may be obviated by the use of sweeps. keleton patterns are chiefly used for pipes and valves, but the method is almost equally applicable for circular covers and other work of a like character. Sometimes, when the choice is between a solid shell pattern and a skeleton, as for valve covers, etc., the skeleton pattern may be preferred because of the ease with which the sand can be rammed between the section pieces, thus enabling a pattern to be moulded by bedding it in the floor, which would otherwise have to be moulded in a box and turned over.

In what follows it is proposed to describe briefly a few representative types of patterns.

A Combined Valve Chest. The combined valve chest (Fig. 26) is a common form of casting. If a first-class pattern were required, the flange timber would be made sufficiently thick to allow of the fillet between the flange and body being cut from the solid timber, while the branches *A* and *B* (Fig. 27) would be turned with prints attached. The simplest method of making such a pattern is by means of sectional construction. Several different thicknesses of timber are required to make up the width over the prints. The rounded parts shown on sections *A* and *B* and the junction between the flanges of section *C* are worked by hand. The centres of each piece are carefully squared over, and the thinner sections screwed to the main section. The method of constructing the

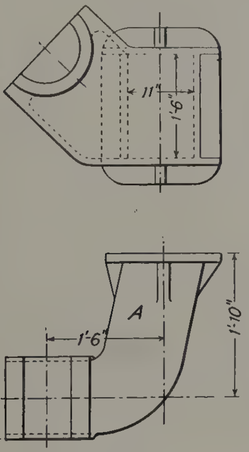


FIG. 28

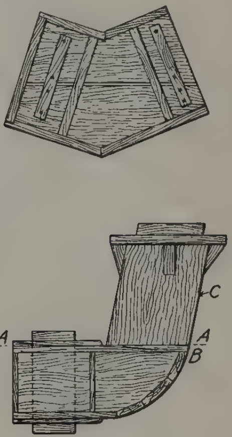


FIG. 29

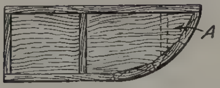


FIG. 30
WYPER SHAFT BRACKET

top branch and flange is apparent from the drawing.

Bracket Patterns. Bracket patterns constitute a different branch of pattern construction. Some brackets of boxed design, having many openings or "windows" through the walls of metal, can be most easily made by coring the entire inside shape of the bracket, and some small delicate brackets must be built up on a block which forms a core print. When a bracket is of very light design and awkwardly shaped, the method of coring should be considered. We shall describe a weight- or wyper-shaft bracket which must be boxed up. One difficulty results from the square section not being all in the same plane, and this difficulty is increased by the large radius shown in illustration. Sometimes the job is simplified by the part *A* (Fig. 28) being quite short—in fact, merely a flange resting on the bottom part—and occasionally there is a large radius, the two sides meeting with the sharp corner rounded off. This is one of the few jobs where the pattern-maker has not to give special consideration to the moulder, but can concentrate his attention on constructing a really strong pattern. An irregular shaped pattern like this must be very strongly made, or a knock in the foundry will render it inaccurate.

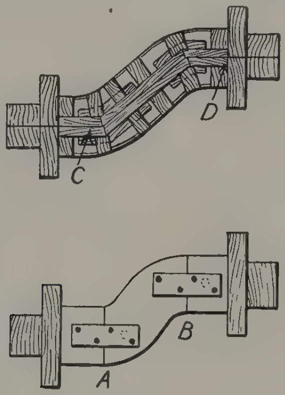


FIG. 31. DOUBLE BEND PIPE

The body of this pattern must be boxed up in two distinct pieces, that below the line *A A* (Fig. 29), forming one piece, and that above *A A*, the other. *B* is a plate which forms the base on which the work is built. It will be observed that the end *A* in Fig. 30 is let into the sides, which prevents any possibility of it being knocked in. The sides have to be in two lengths, butted together. With regard to the thickness of timber, it should not be less than $1\frac{1}{4}$ in. The large radius would be staved as shown. This completes the boxing-up of the bottom part of the pattern, and boxing the top part *C* in Fig. 29 is comparatively easy. The shape of the sides is obtained from the full-sized drawing, and they should be checked at the ends, so as to form a shoulder for the end pieces. The wyper-shaft bracket pattern described might be moulded with the square flange down, the moulder either making a joint round the large radius and down to the top of the square flange, or carrying it along the line *A A* (Fig. 29), in which case the square box would have to be screwed, so that the moulder could loosen it in the sand and draw it out after the bearing piece had been withdrawn.

Double-bend Pipes. For what are called double-bend pipes the skeleton method is very suitable. Fig. 31 shows one such pipe. In one sketch the pattern is shown unfinished to illustrate the plate construction. The most practical way of making an awkward pipe such as this is to set down a plan on a level floor or building board, and erect faces. The flange centres would then be marked on these, and the bottom halves of the flanges screwed to them. The centre-plate would be broken, as shown at *A* and *B*, and strong bars, cut to the shape, screwed at the back. A ground should first be screwed to the flange and kept to the thickness of the plate below the centre. The centre-plate should now be proceeded with. It must be kept broader, and the S-shape drawn on it after it is finished. The end pieces, *C* and *D*, of the centre-plate can now be screwed to the ledge, formed by the ground on the flange, and a bracket placed at the end of it. The centre portion of the plate should now be fitted between the two parts and the bars screwed on. The S-shape can now be drawn on, and the whole plate taken away and cut to the proper shape. When the plate has been replaced, the grounds can be fixed at the short angle formed by the joining of the plate. The grounds will have to be "doctored" in to sweeten the curves. The sketch will make this clear.

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XIV

THE PLANING MACHINES—(*contd.*)

Methods of Driving. Variations are noticeable in planing machine drives, similar to those which are adopted in other tools. In by far the largest group yet, a belt drives from a countershaft, or from an electric motor located on the machine, while the regular details of dog and tappet reversals, and the later changes of speeds through gears are retained. Yet the all-electric drive grows in favour, being eminently adaptable to rapid changes in speeding, and in quick reversals, since these are under direct electrical control, nstead of by the slower action of dogs on the moving table.

Examples. Figs. 95 to 99 are illustrations of the planers by Smith & Coventry, Ltd., Manchester, made in a range from 24 in. by 4 in. to 72 in. by 72 in. The drawings are hose of the 30 in. by 30 in. machine, taking 6 h.p. to drive it. These are driven with belts from the line shaft with fast and loose pulleys as shown, or an electric motor can be substituted on the countershaft on top of the machine, or a motor with a three-speed gear box can be supplied. The cutting speed provided is 40 ft. per minute, and the return is 140 ft. per minute in the smallest machines, and 60 ft. per minute in the largest. Feeds range from $\frac{1}{32}$ in. to $\frac{1}{2}$ in.

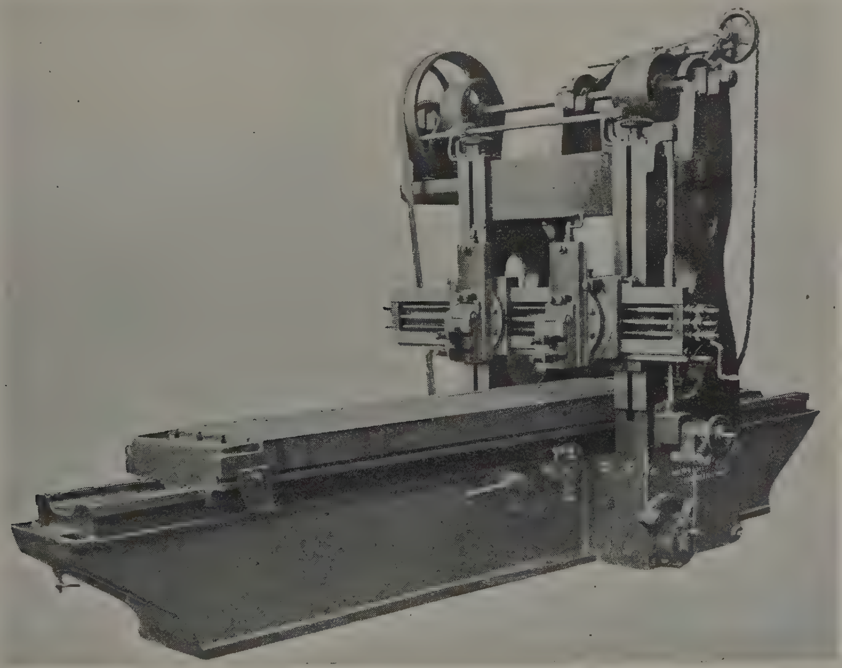


Fig. 95. 30 in. Planing Machine

The photograph, Fig. 95, and the general drawings, Figs. 96 and 97, show the deep bed and table, the semi-parabolic housings tied at the top by a deep cross brace with an independent platform to carry the fast and loose countershaft pulleys with striking gear operated by a dependent chain, and at the opposite end the two pulleys of different diameters that drive, one with a crossed belt to large pulleys below, for cutting, the other with an open belt to small pulleys for reversing. The cross rail, elevated with screws and bevel gears, carries two tool

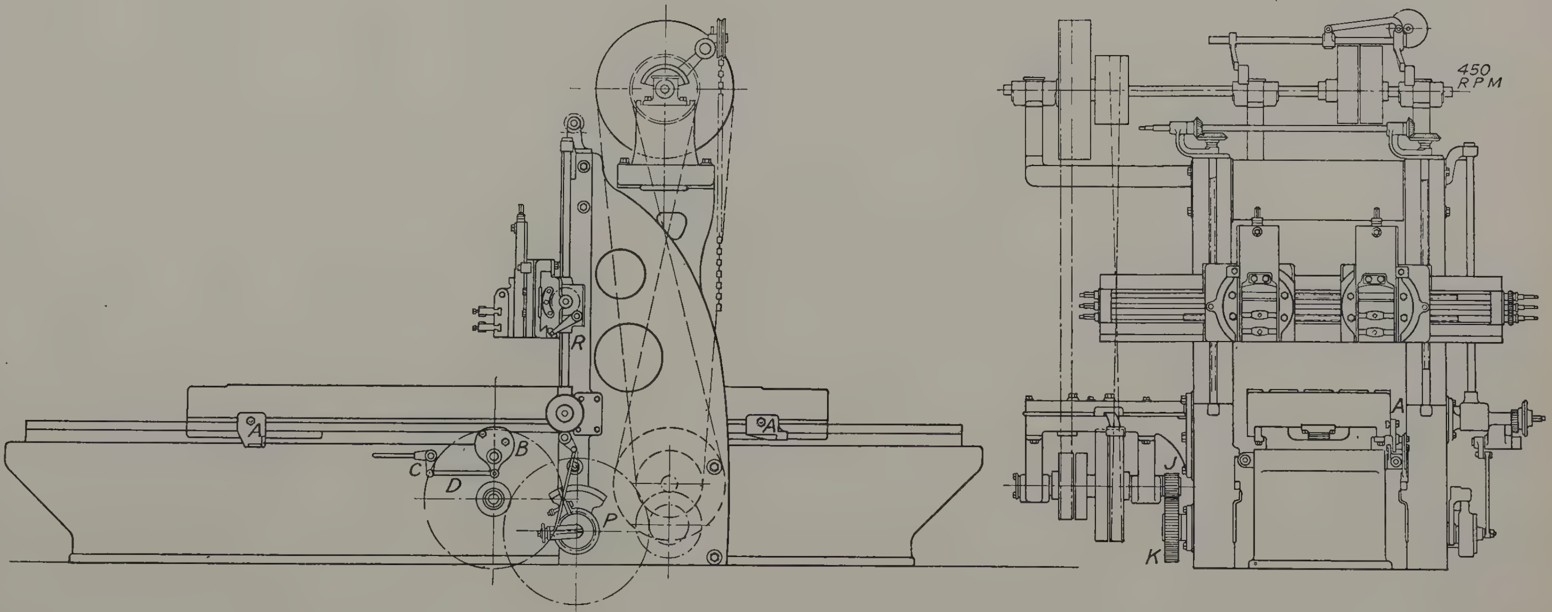


Fig. 96. General Views of Planer

boxes with their saddles lengthened on their outer ends. Their screws and splined rod are seen, with square necks for hand adjustments along the cross slide.

The Mechanism. This can be traced out with the help of the detail drawings. In the general and detail views, Figs. 96 and 97, the dogs *A*, *A* strike a tappet disc *B*, seen in section in Fig. 98, the contacts being made against rollers. The shaft *C* to which the rocking of the disc is communicated through the horizontal rod *D* and levers passes through the bed, Fig. 98, which permits of making hand adjustments of the table stroke from either side. The connection to the striking gear at reversals is indicated in Fig. 98, but in enlarged details in the associated group, Fig. 99. On the farther side of the bed, a rod *E*, Fig. 99, actuated by the rocking of the shaft *C* through a short lever, Fig. 98, communicates motion through a lever to a vertical rod, *F*, at the upper end of which a lever pulls over the cam plate *G* for changing the belts on adjacent pulleys from fast to loose and vice versa, the movement pivoting round *H*. The effect is to draw the plate *G* along on the casting that houses the pulleys, and move the short levers on the striking forks through the dog leg cams. One of these forks is shown in plan—that for the large cutting pulleys. The movement of the short arms in the cams is slight, that of the forks is ample, rapid, and simultaneous, and one belt is removed on to the loose pulley before the other is brought into action. The loose return pulleys, Fig. 97, have heavy rims to store up energy for the reverse.

The transmission to the table is as follows: The pinion *J*, Figs. 97, 98, on the reversing shaft drives a gear *K* on a shaft that runs through the bed. This one carries a 33 toothed pinion *L*, meshing with a wheel *M* on the shaft of which a pinion *N* drives the 75 toothed bull wheel that engages the rack. Being so large, seven of its teeth are always in mesh. The same train of gears drive and reverse, the difference in speeds being provided for by the variations in the sizes of the pulleys on the countershaft, and in those of the pairs of pulleys on the machine. The feed disc *P* is keyed on the intermediate shaft, and is connected through a lever with a quadrant gear and pinion *Q* and mitre gears to the vertical rod *R* to the screws and splined rod on the cross rail. An examination of the drawings will reveal many details of construction which do not call for explanation.

In a planer by Joshua Buckton & Co., Ltd., of Leeds, motor-driven, the motor of constant speed type is mounted on the cross girder of the machine, and is coupled direct to a three-speed gear box, giving cutting speeds of 30, 45, and 60 ft. per minute, with a constant quick return of 90 ft. per minute, the driving being with belts from the countershaft. The table is driven with a spiral pinion meshing at an angle of 45° with the teeth of a square rack. By means of a special feed gear, quick traverses are obtained by the simple holding out of a lever, and the feeds of the various boxes are independently reversible, and variable over a wide range. The firm now builds a machine with a reversing motor drive, and fitted with a dial control that dispenses with the table dogs. The dial is geared to the table drive, and reproduces the motion of the table on a reduced scale. The dogs having no heavy work to do, as they merely throw the reversing switch release to the feed gear, are comparatively light. The feed gear is either driven off the motor generator set that forms part of the reversing motor equipment, or by a separate small motor.

A mechanism that has been applied to several hundreds of planers and slotters is the "Vulkan" reversing electro-magnetic clutch, applicable both to driving and feeding. A central disc armature shrunk on the shaft is flanked by two electro-magnetic bodies running loose on the shaft, each being fitted with an annular energizing coil, each body having a ventilating ring to dissipate heat generated. The disc armature is fitted with a wooden friction ring. When continuous current is supplied to one of the coils through slip rings, the loosely running magnetic body compresses springs and advances towards the disc armature, a distance of about $\frac{1}{16}$ in. bringing the ventilating ring in contact with the friction ring and so driving the shaft. To reverse, the current is switched by an automatic switch over from one energizing coil to the other. It will reverse accurately at both ends of the stroke, and on strokes of a few inches only. It is applied to feed motions, as well as to table drives.

The Lancashire Drive. This has had the most powerful influence in recent years on planer machine design. It consists of a motor generator set, arranged for either direct or alternating current. The motor is coupled to the table through gears. Speed variations, and reversals are obtained by controlling the speed, and the

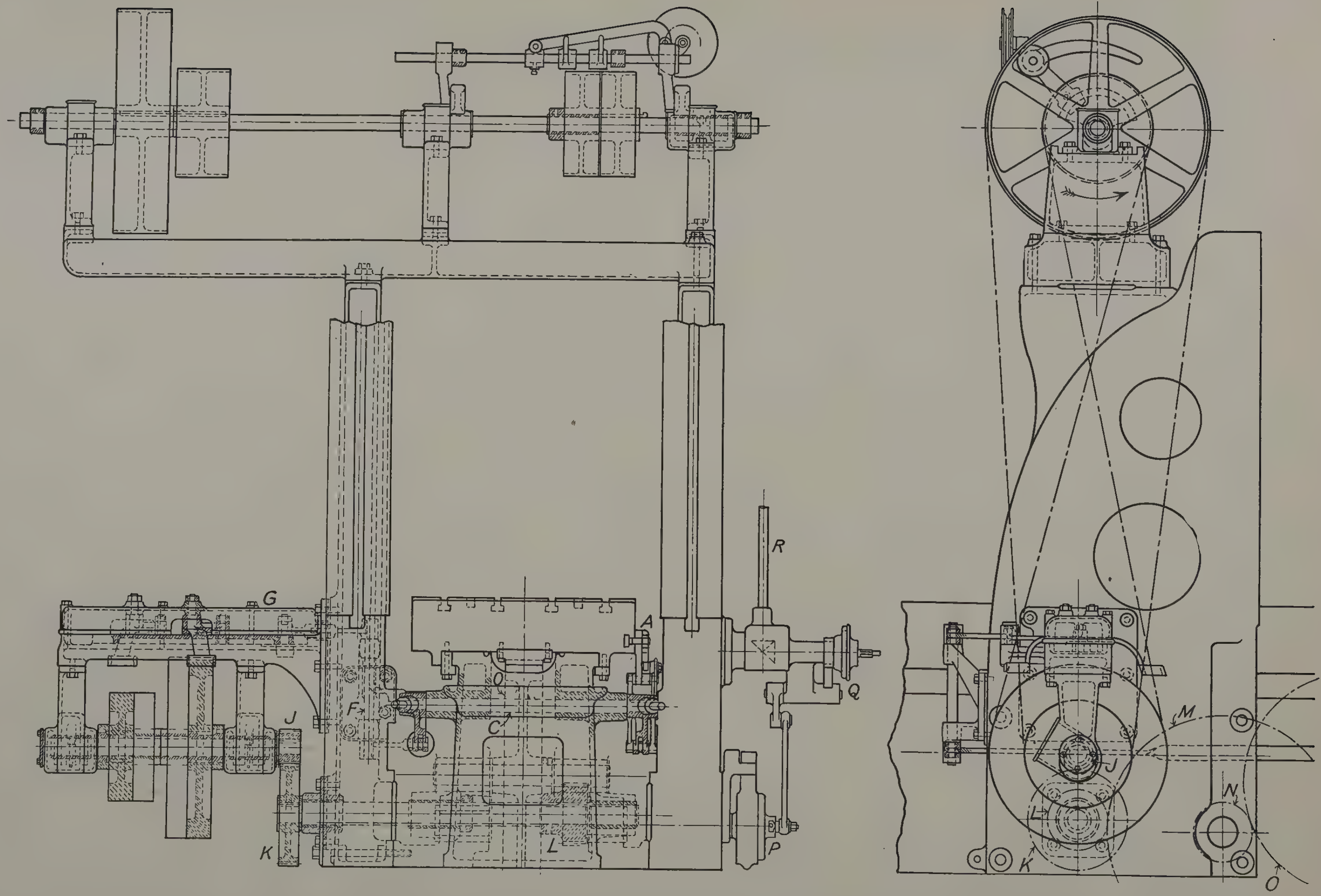


FIG. 97. PLANER DETAILS

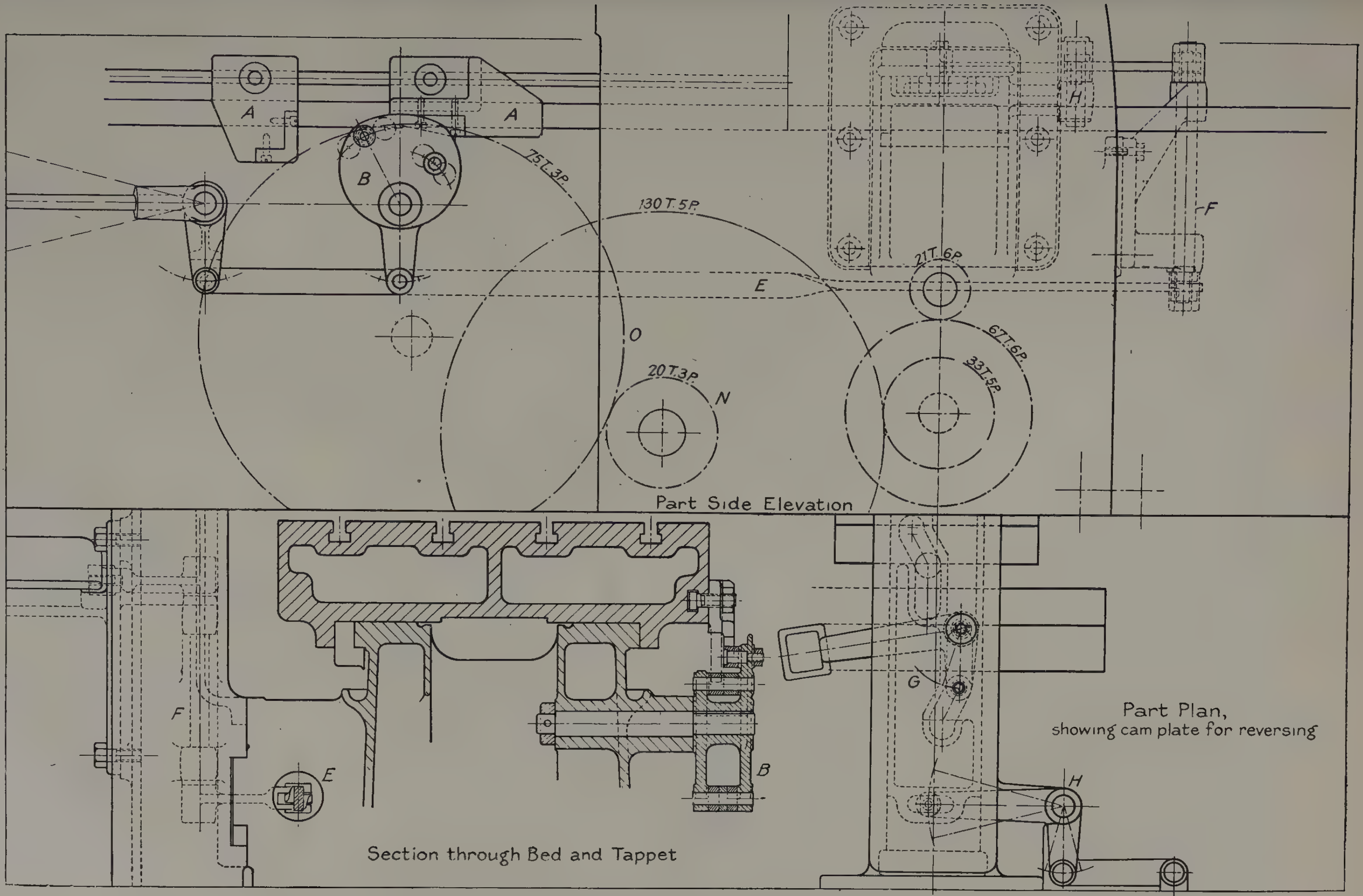


FIG. 98

FIG. 99

REVERSING GEAR OF PLANER

direction of rotation of the motor, and these are governed by the voltage, and the direction of current from the generator. An accelerating switch when fitted, quickens the return speed between cuts when two or more surfaces at some distance apart are being planed. It also increases the cutting speed after the tool has entered its cut, so reducing the shock and stress on the tool point.

John Stirk & Sons, Ltd., of Halifax, have specialized in machines that have achieved a remarkable record of efficiency and range. Fig. 100 is a striking photograph of a small machine of 24 in. by 24 in. by 6 ft. capacity on the table of one 12 ft. 6 in. wide. The first is a belt-driven type, the second is electrically driven. The small planer type—the "Veloplane"—is made in capacities from 24 to 72 in. with a stroke of table from 6 to 16 ft., to be extended if desired. One, two, or four tool heads are fitted, two in the latter case being on the housings. The drive is by open and crossed belts on the pulleys of different sizes on the countershaft, to pulleys of equal sizes below. In the simplest design cutting speeds are single, ranging from 48 ft. per minute in the smaller, to 33 ft. per minute in the heavier. The rate of return is uniformly 100 ft. per minute. In a modified design the drive is by open belts to larger and smaller pairs of pulleys, but a gear-box is provided to give two or three cutting speeds from 24 to 50 ft. per minute with a constant return of 100 ft. per minute. In another variation the drive is with an electric motor mounted on the top girder. The machine being thus self-contained, can be set down in any position. And either a constant speed motor is fitted with geared changes of speeds, or a variable speed type furnishing any cutting speed between 25 and 50 ft. per minute, with constant, or variable return speed up to 100 ft. per minute.

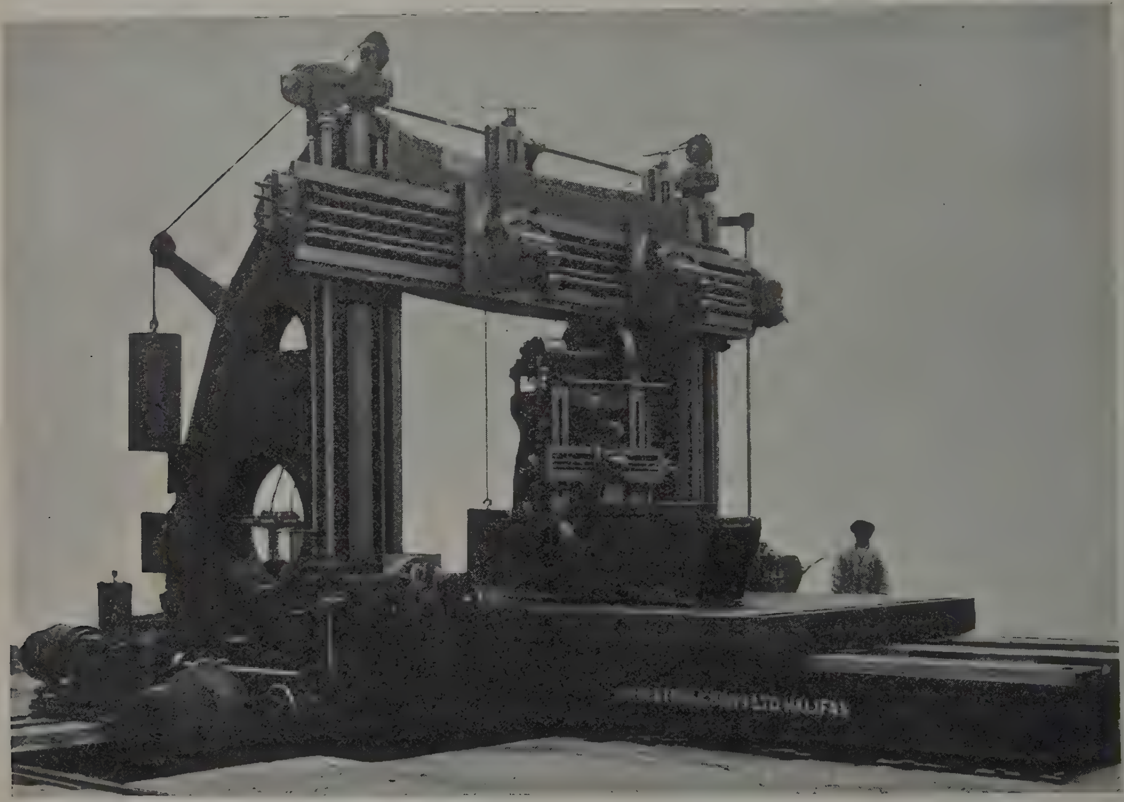


Fig. 100. A Small Planing Machine on the Bed of a Large One

The most remarkable type of machine built by Messrs. Stirk is the "Hiloplane," embodying the Stirk patent "split field" reversing motor equipment. The term denotes the division of the shunt field windings of both generator and motor into two sections. In the case of the generator, these sections are connected to oppose each other, so that alternative polarity results from the alternative excitation of the two sections. The two sections of the motor field are connected to agree, and one is permanently excited. The second section is excited during the cutting stroke only, and a slow cutting speed, and a quick return are thereby provided.

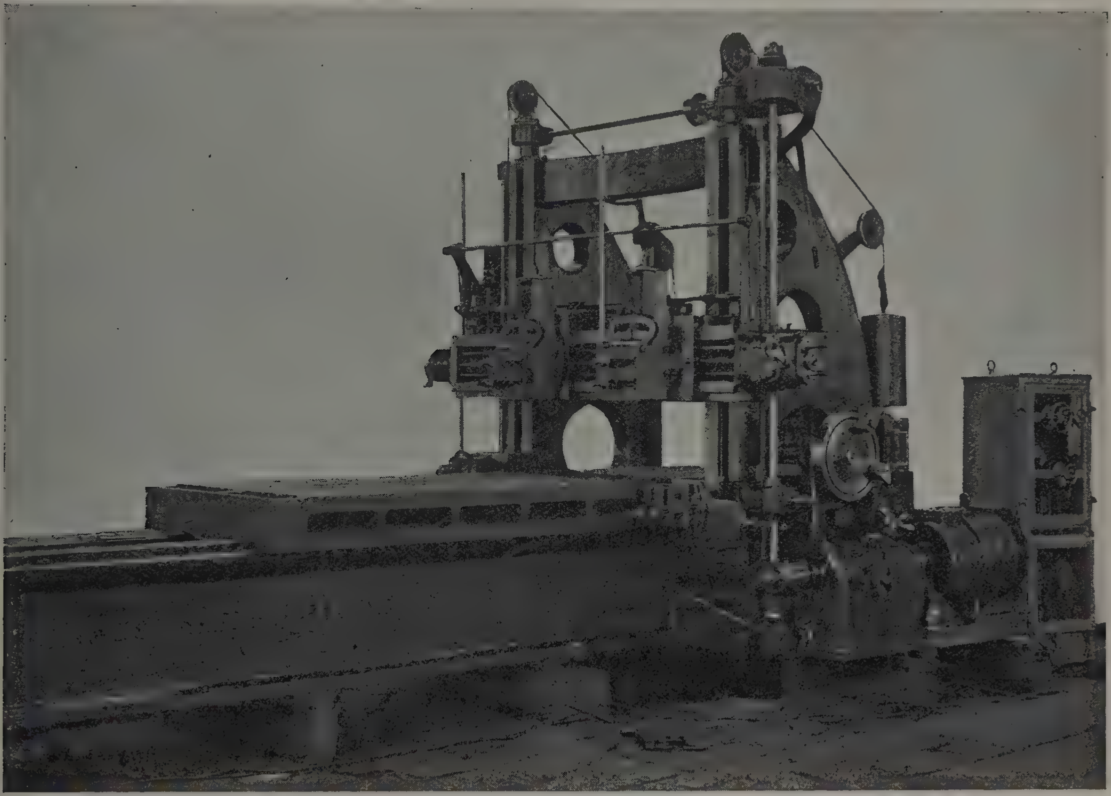


Fig. 101. The Hiloplane

The Stirk design comprises a motor coupled direct to the gearing, supplied with current from a motor generator, either on direct or alternating current. By manipulation of its magnetic field the generator may be made to generate at variable voltage, and alternate polarity. The final motor being independently excited, the variation of generator voltage gives variable speeds, and the alternation of polarity causes a corresponding reversal of the direction of rotation.

A photograph of the 4 ft. by 4 ft. by 12 ft. machine, electrically driven, is shown by Fig. 101. These are built up to a capacity of 14 ft. 6 in. wide. In the largest machines the bed is cast in three sections, tongued and bolted together, and three bearing surfaces are provided for the table. The table is also cast in sections, and is hollow. It is driven with twin racks cut from solid steel. The smaller sizes have a single rack, and two sliding ways. Another feature is that wheels are not keyed on their shafts, but are cut solidly with them to enable them to stand the high speeds and rapid reversals.

In Fig. 101, a clear view is seen of the disc and its master control switch by which the stroke of the machine is regulated without table dogs. The dogs on the disc can be set to give a definite length of stroke, and tightened. An inner circle of graduations on the disc provides for cross and vertical planing movements, which is another special feature of the machine, permitting of dealing with work that has facings and bosses on its ends that would without these movements require a separate setting.

The feed, which is of the Stirk patent electric quick change type, is of special interest. Eight or more positive feeds from $\frac{1}{32}$ in. to $2\frac{1}{2}$ in. are regularly employed. They may be changed or reversed instantly, and are clearly marked for the convenience of the operator. There is no waste stroke for the feed, which is virtually instantaneous, and the accuracy is that of a dividing mechanism. The same motor that goes step by step for feeds can be run continuously for quick and fine power traverse, the control for this valuable motion being by a double-ended pendant switch, easily held in the hand.

JIGS AND TOOLS

By Leo Kelsey, A.M.I.P.E.

LESSON VII

AUTOMATIC TURNING MACHINES

Some observations have already been made on automatic turning machines and their essential functions. This subject is so important to modern tool engineers that we shall go farther into the explanations of this class of machine tool.

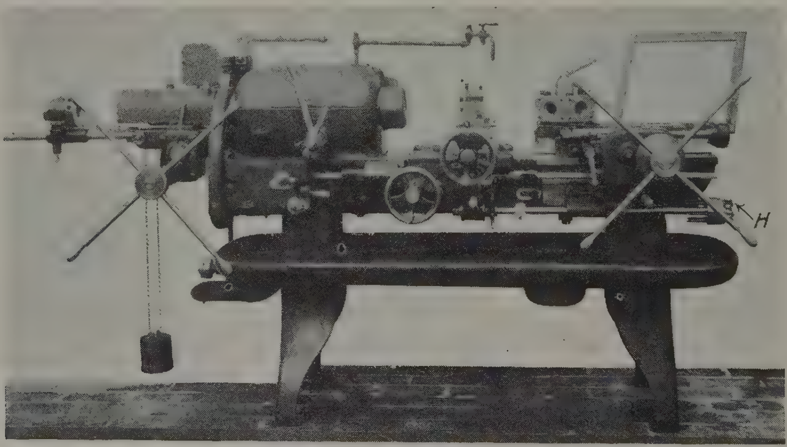


Fig. 52. No. 4 Capstan Lathe
(Messrs. Alfred Herbert, Ltd., Coventry)

Referring to the illustration, Fig. 52, this tool is the No. 4 capstan lathe, manufactured by Alfred Herbert, Ltd., Coventry. The saddle supporting the capstan or turret head is traversed by means of the hand-wheel shown. Each time this hand-wheel is rotated in a clockwise direction the turret is moved away from the work. Further rotation of the handle causes the turret to revolve, thereby bringing a new tool into line with the work. The stock is fed through the spindle, which is hollow, and new stock is fed automatically upon the completion of any one piece. Threads can be cut on these machines, and automatic feeds to capstan slide are provided. The feeds are put in and out of engagement by the stops marked *H*, which are also used as dead stops for the component being machined.

Automatic Screw Machines. There are many

I

Component

Top Roller and Casting.

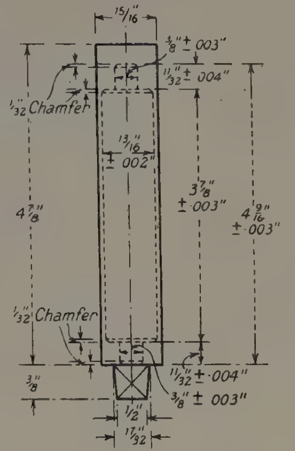


Fig. 55.

II

Spindle 1.

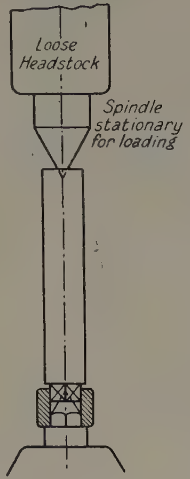


Fig. 56.

III

Spindle 2.

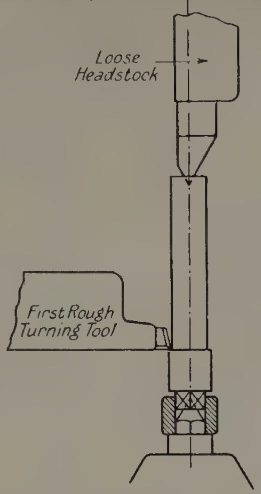


Fig. 57.

IV

Spindle 3.

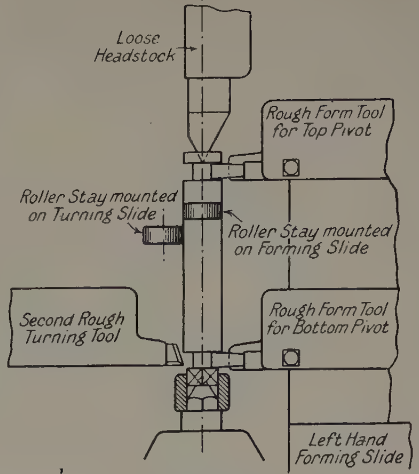

Loose Headstock
Rough Form Tool for Top Pivot
Roller Stay mounted on Turning Slide
Roller Stay mounted on Forming Slide
Second Rough Turning Tool
Rough Form Tool for Bottom Pivot
Left Hand Forming Slide

Fig. 58.

V

Spindle 4.

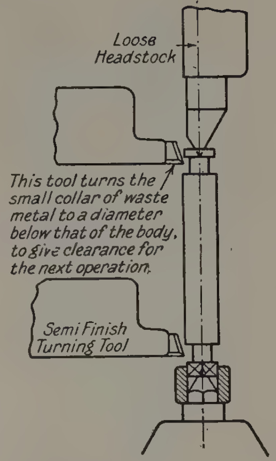

Loose Headstock
This tool turns the small collar of waste metal to a diameter below that of the body, to give clearance for the next operation.
Semi Finish Turning Tool

Fig. 59.

VI

Spindle 5.

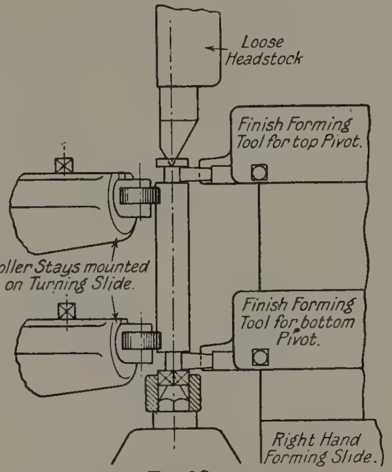

Loose Headstock
Finish Forming Tool for top Pivot.
Roller Stays mounted on Turning Slide.
Finish Forming Tool for bottom Pivot.
Right Hand Forming Slide.

Fig. 60.

VII

Spindle 6.

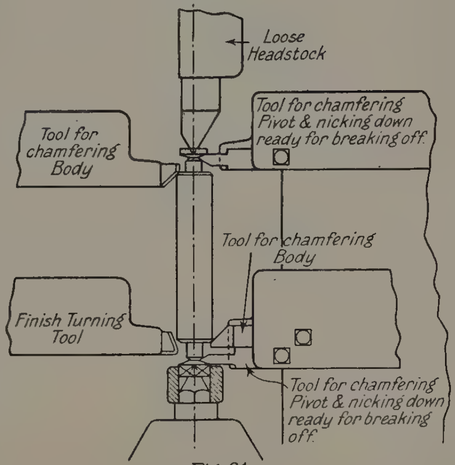

Loose Headstock
Tool for chamfering Body
Tool for chamfering Pivot & nicking down ready for breaking off.
Tool for chamfering Body
Finish Turning Tool
Tool for chamfering Pivot & nicking down ready for breaking off.

Fig. 61.

Tool Layout for Vertical Automatic Turning Machine classes of these machines on the market, and they all have the same characteristic, viz., that operation of the tools is accomplished by means of suitable cams mounted upon drums, which are connected by gears to the machine spindle. The action of these machines, when once they are set up, is automatic, the operator only having to feed new stock.

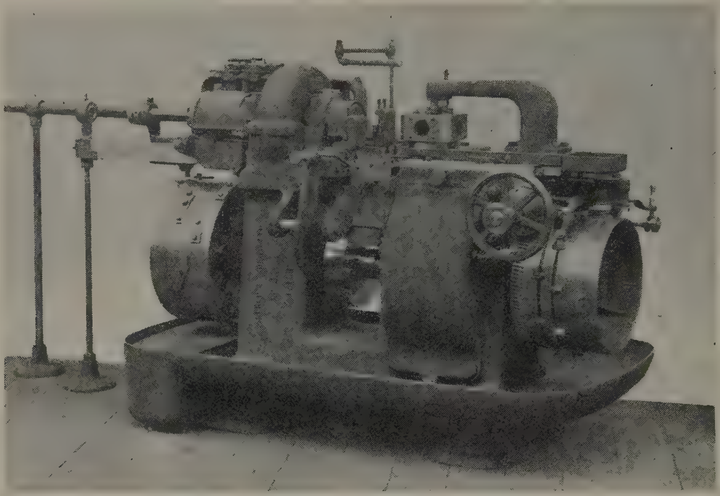


FIG. 53. AUTOMATIC SCREW MACHINE
(Messrs. Alfred Herbert, Ltd., Coventry)

A typical tool of this class is shown in Fig. 53, which shows an automatic screw machine by the same maker as the capstan lathe. These machines are capable, when properly set up, of exceedingly high output, the limit in many cases only being reached when the actual strength of the component being machined is also reached.

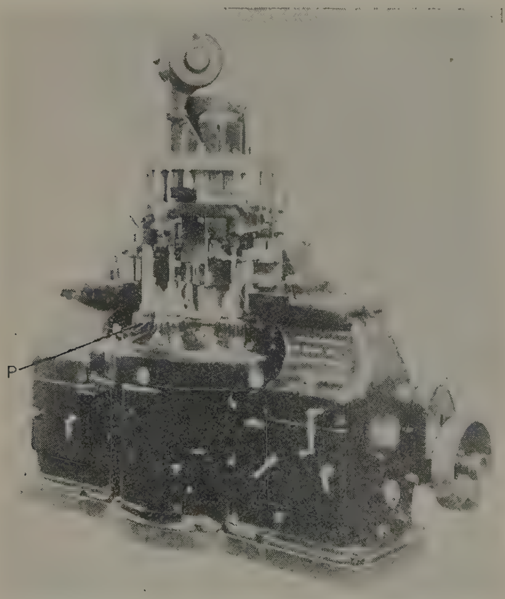


FIG. 54. VERTICAL AUTOMATIC TURNING MACHINE
(Messrs. T. Ryder & Son, Ltd., Bolton)

The Vertical Automatic Turning Machine, an illustration of which is Fig. 54, is manufactured by T. Ryder & Son, Ltd., Bolton, and is used for turning and forming components which have been centred or are held in the chuck.

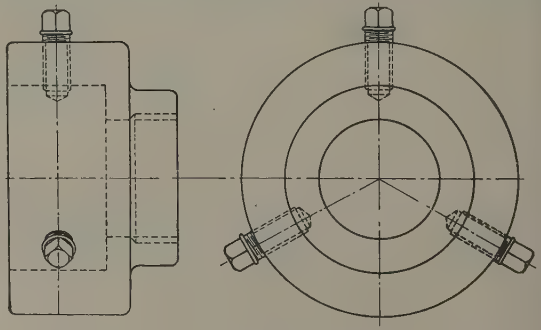


FIG. 62. BELL CHUCK

Six rotating work-holding heads are arranged in a circle, and indexed, causing them to come under different tools, etc., successively. The heads have no vertical movement, the various tools, etc., sliding up and down the central pillars, when operating upon the component. There are only five tool heads on the pillar, thus leaving one work-holding head free for the operator to remove the finished part and insert a fresh blank. A tool layout for the machine is shown in Figs. 55 to 61. The cut shows the machine set up for this component. Tooling on automatics largely depends upon the make of machine, and the subject must be studied in detail from this aspect.

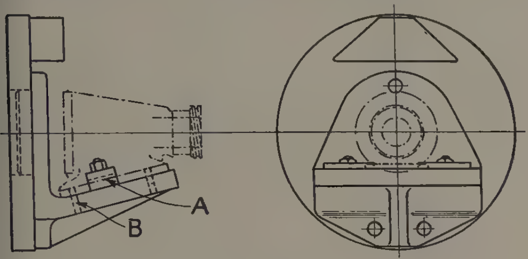


FIG. 63

Turning Fixtures. In Fig. 62 is shown a bell chuck; this is used for components of awkward shape, which are held by the screws. The chuck is screwed to suit the nose of the lathe it is to be used on.

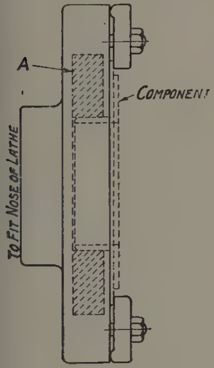


FIG. 64

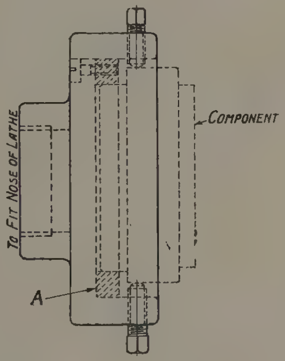


FIG. 65

Fig. 63 is a fixture for holding the casting shown. The pins *B* locate the component which is held by the clamps *A*. This type of fixture is very useful for machining components which have been machined on a face which is at an angle to the axis of the hole or part which is to be turned, screwed, bored, etc.

In the next, Fig. 64, is shown a jig for machining castings having a round spigot from which to locate. This type of fixture is easily adaptable, by means of changing the ring *A*, to locate work of fairly wide dimension variation. There are two forms of the jig shown in the illustration, one of which (Fig. 65) is an adaptation of the bell chuck, and the other of the detachable clamp variety, which is widely used for machining thin, springy components, such as bevel gear blanks, packing discs, etc., which are difficult to hold in an ordinary three-jaw chuck without distortion.

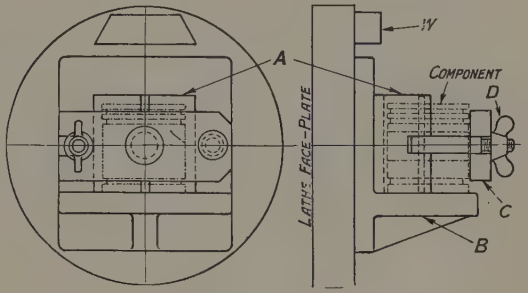


FIG. 66

The boring jig shown in Fig. 66 is for boring and reaming the gudgeon pin-hole in automobile pistons. In many plants, this operation is performed in drilling machines, and a suitable fixture is provided, but where the output is not high and the degree of accuracy required reasonably close, the lathe provides a very economical method of achieving this. The piston in the illustration is located by means of the vee-block *A* and the angle-plate *B*, and secured by the clamp *C* and the nut *D*. The clamp has a hole larger than the hole to be bored in the piston, which clears the various boring tools, cutters, etc. As a rule, machining the inside face of the gudgeon pin bosses is not done in this operation. This fixture is provided with a counterweight *W*. Sometimes the

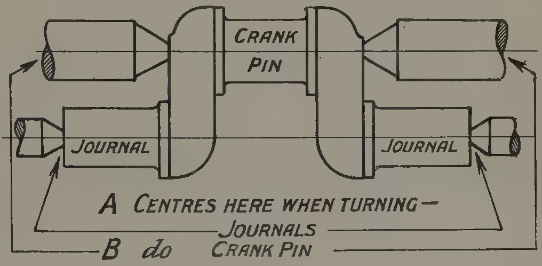


FIG. 67

angle-plate *B* is formed integral with the face-plate and screwed on to the nose of the lathe. However, if the quantity is very small, an ordinary angle-plate with the vee-block secured to it is quite satisfactory, although somewhat slower to set up.

It should be noted that, where the quantity does not warrant the installation of a special boring machine, it is quite possible for the majority of lathes to be converted to perform this function. When using the lathe as a boring machine, the work or component is clamped to the saddle and the cutter fixed in the nose of the lathe. If necessary, the cutter can be supported by the poppet-head centre, and the work traversed towards the cutting tool. However, the operation is not a fast one, and if there is a likelihood of a repetition of such work, a special boring machine should certainly be used.

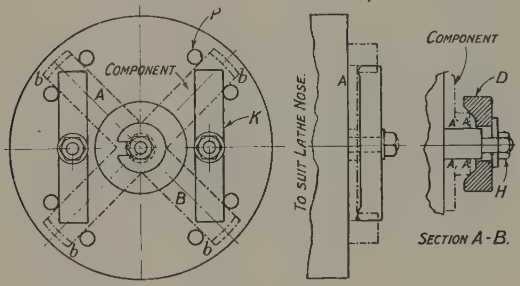


Fig. 68

Crankshaft turning is done in the lathe by first turning the journals upon centres, as at *A* in Fig. 67, and then turning the crankpin upon special extensions of the centre, as at *B* in the same illustration.

The turning fixture for machining worm segments is shown in Fig. 68. These segments have been machined on the surfaces *A*, and it is required to turn their large diameters *b*. This is accomplished in the jig shown, by loosely locating them on the pins *P* and pulling them on to the face *A* by means of the taper clamp ring *D*, which is tightened by the nut *H*. The clamps *K* serve to press them on to the face of the fixture. In the fixture shown, four only are machined at one setting, but should there be a large quantity to machine, more can be accomplished at one setting by suitably placing them in the fixture.

The simple turning fixture in Fig. 69 is used for cutting the thread in the worm nut shown. This fixture is bored and screwed on its boss to fit the nose of the lathe on which it is to be used. The component is secured by means of the swinging clamp *B*, and the screw *H* presses it against one side of the fixture, which locates the component in relation to the cored hole *F*. The swinging clamp *M* presses the component against the locating face *W*. This is done in order that the face *K* of the component shall be in correct relation to the screw cutting tool, thus ensuring that the thread to be cut is in correct relation to the face of the component. This fixture is made from a one-piece casting, to ensure rigidity whilst the worm is being cut, as this is a fairly heavy operation. The fixture just described is frequently used in thread-milling machines, which provide a rotating cutter in place of a stationary tool as used in a lathe.

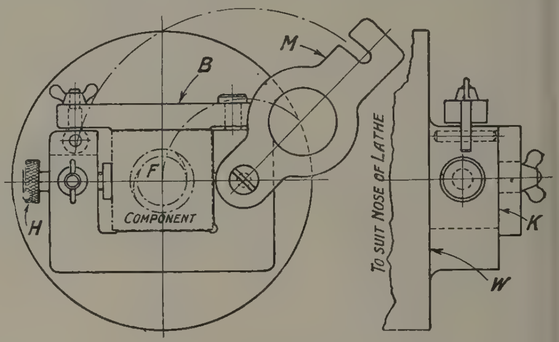


Fig. 69

GAUGING AND INSPECTION

By Joseph G. Horner, A.M.I.Mech.E.

LESSON II

FIXED GAUGES

While the methods last described are adopted in the general and repair shops, they are supplemented or superseded by more precise systems when components are manufactured in large quantities to interchange. Here the fixed gauges are used, and micrometric and vernier instruments.

Fixed Gauges Compared. These are broadly divisible into cylindrical and snap; the first being as denoted by their name, the second being flat and thin with small areas of contact. The first fill a hole or encircle a shaft, the second measure only a narrow zone. The first are used less than formerly, the second more frequently. The reason is, that the surfaces in contact being larger in one case than in the other, the test is not quite so refined. A snap gauge will detect a variation that a cylindrical one will not show. This is in harmony with a fact previously noted, that narrow points are to be preferred to those that are wide for calipers. These are fine distinctions, but the function of the fixed gauges is to deal with these. $\frac{1}{1000}$ in. reads like a fine dimension, but in fact it is very coarse. A difference of that extent in a shaft and its bearing would be a slack fit. That is required in many instances, but in the modern system of gauging, its amount is not left to be determined by the workman, but is embodied in the gauges which provide the exact variation from the absolute dimensions to suit any kind of fit desired.

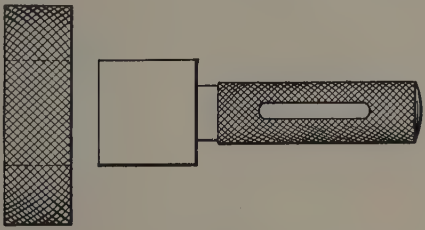


Fig. 6. Plug and Ring Gauges

The Meaning of Limits. Working to limits means the determination beforehand of the dimension to which a piece of work shall be finished, in order that it shall be suitable for the purpose for which it is to be used. Obviously, this must vary widely in the case of different classes of work of finer or coarser grade. It may range at extremes from $\frac{1}{50000}$ in. to $\frac{1}{500}$ in. with a big difference in the cost involved. Economy requires that limits shall be strictly selected to suit the kind of fit that is desired or necessary, since it is expensive to insist on the observance of fine limits where they are not wanted. The limit system is not to be confounded with the mutual fitting of one part to another, for it ensures interchangeability which is impossible with the other. Four kinds of fits are provided for between shafts and their holes. *Force fits* in which the shafts have to be forced into their holes with hydraulic pressure, or, the holes have to be expanded with heat to embrace the shafts. *Driving fits* mean that shafts have to be driven into their holes. *Push fits* mean that shafts can be pushed into their holes, but will not be free to rotate. *Running fits* are divided into three grades—the most easy for engine and other work, another for high speeds and good average machine work, and the third for the finest tools. In this system it is usual to make the hole the basis of measurement, and to confine the limits to the shaft or spindle. This is because less difficulty is involved in tooling a shaft to size than in the case of a hole. It is not a very serious matter, where adjustable reamers and grinding wheels are used, and measurement is done with fixed gauges. Both systems are provided for.

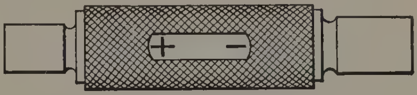


Fig. 7. A Limit Plug Gauge

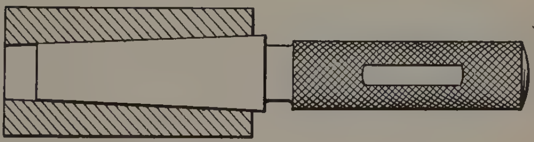


Fig. 8. Plug and Ring Taper Gauges

Tolerances. As it is not practicable to work to absolute limits, drawings are now marked, and gauges made with tolerance allowances, that is, clearly defined departures that are permissible from the exact limits. These are fine enough to permit of the interchangeability of parts, and they do not make too severe demands on the tooling operations. In this system the workman does not measure, but gauge, the limits and tolerances being included in the gauge.

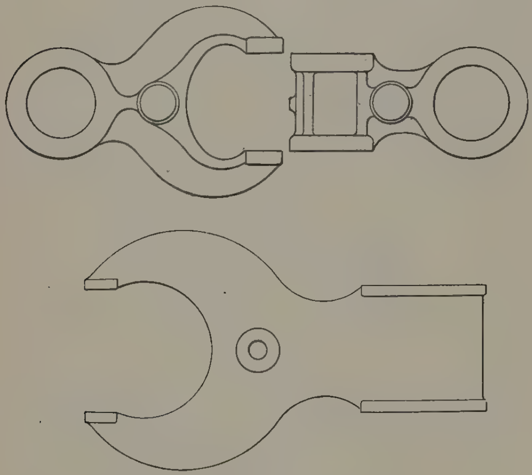


Fig. 9. Forms of Snap Gauges

Cylindrical Gauges. The early forms of these, plug and ring (Fig. 6), were lapped or ground as nearly as possible to exact sizes. They are so made now, within a limit of accuracy to nominal size of 0·0001 in. They are used much less in actual operations than the limit gauges of the snap type. If cylindrical gauges of the same size are slid one over the other without oil, and allowed to remain a short time, they will seize, and separation will be difficult. But if a thin film of oil is rubbed over them with the finger, they can be inserted and removed with a gentle twist of the hand. When these are made to limits, the plug (for the bore) is double-ended (Fig. 7), with a central milled-grip marked − and +, or "go in," and "not go in," the latter being often made shorter than the former to enable the workman to see the difference at a glance. Cylindrical gauges are also made for tapers (Fig. 8).

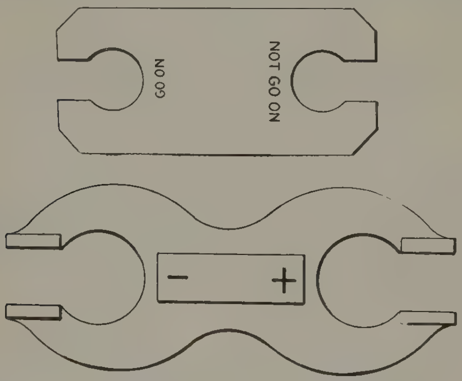


Fig. 10. Forms of Limit Snap Gauges

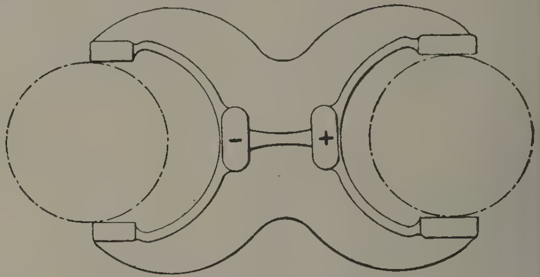


Fig. 11. Horseshoe Limit Gauge

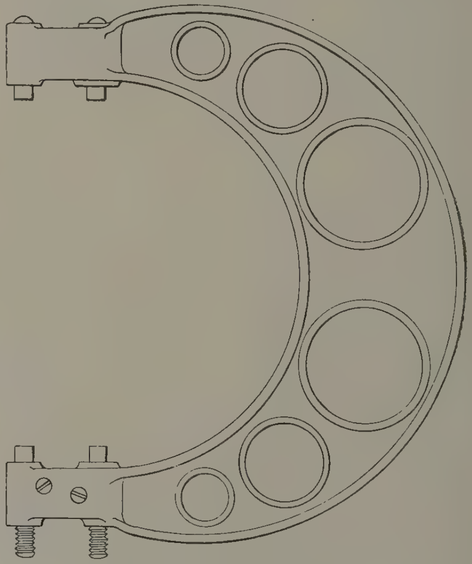


Fig. 12. Horseshoe Limit Gauge, Adjustable for Both Limits

Snap Gauges. These are made in several outlines, internal and external (Fig. 9), separately with handles, or on opposite sides of the same bar (Fig. 9), and made also either exact within from $\frac{1}{5000}$ in. to $\frac{1}{10000}$ in., or to limits (Fig. 10). They are drop forgings with hardened faces, ground and lapped. The horseshoe (Fig. 11) is a variety of the snap gauge, suitable for work of larger diameters than the plate gauges take. When above about 2 in. it is more convenient to have the "go on," and "not go on" points on the same instrument (Fig. 12), and to make provision for their adjustments to give a range of limits, while the hole remains fixed at standard. This is a great concession to firms, that, requiring several limits for different classes of fits, avoid the accumulation of an extensive stock of fixed gauges by employing the adjustable forms. The Newall Engineering Co., Ltd., of Warrington, have developed this system with notable success.

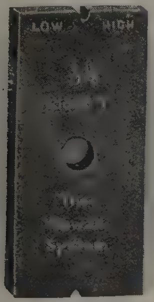


Fig. 13. A Limit Reference Bar

The Newall System of Limits. These are based on the hole, and include the four classes of fits previously stated, each with tolerances that vary with diameters. And the tolerances are supplied in two grades, higher and lower for the adjustable points. They are made in sets with capacities from ½ in. to 30 in., all above 6 in. with increments of 1 in., smaller sizes from ¼ in. to ½ in. Two methods of setting the points are adopted. One is by means of limit reference bars, which are ground to the high and low limits of the particular diameter and class of fit. Each bar (Fig. 13) is ground cylindrically on the ends, and guaranteed to size within 0·0002 in., or 0·005 mm., giving the high and low limits. The other method is to set both points on a standard reference bar, that is one ground on the ends within 0·0002 in. of standard, and then set to the limits required by means of dials (Fig. 14). These have taper shanks that fit into taper holes provided in the screws of the gauges. Having set both points of the gauge to the nominal size obtained from the standard reference bar, the screws of the gauges are locked and the dial and index are inserted in the holes of the screws at zero. The screw that carries the dial is then released, when it can be adjusted to the allowance, either above or below standard, by rotating the screw in the plus or minus direction, as shown by the arrows on the index. The screw is then locked, the dial and index reversed, and the process repeated with the other point of the gauge. One pair of dials suits all sizes of gauges up to 6 in. The main divisions are in thousandths, and the subdivisions, quarter-thousandths. A hole is provided in the frame to receive buttons that denote the class of fit to which a gauge has been adjusted. The Newall Co. provide printed sheets to be hung in workshops, giving all the limits and tolerances for each diameter and class of fit that are included in their very extensive system.

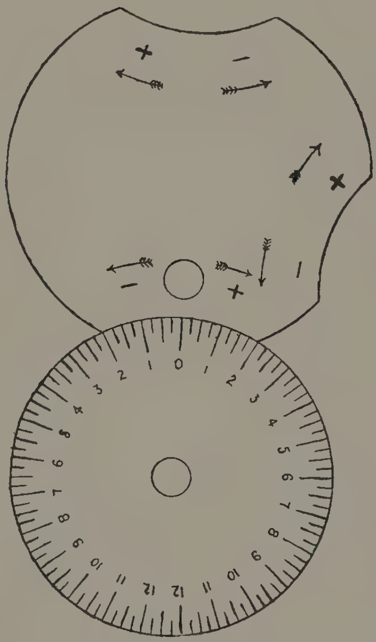


Fig. 14. Dials for Making Adjustments for Limits

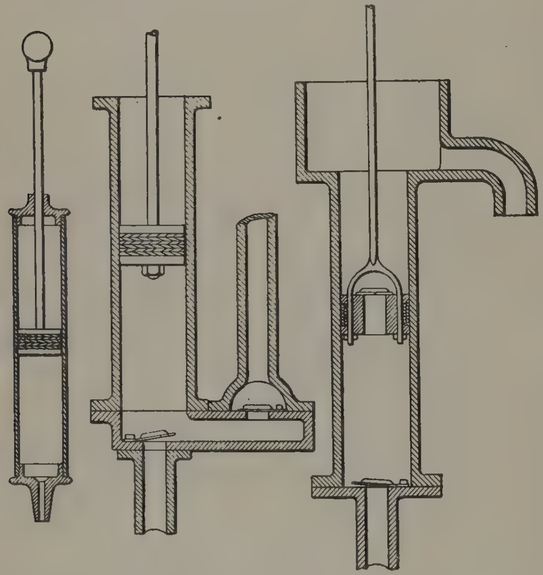
NOTABLE EVENTS IN ENGINEERING HISTORY

By E. A. Forward, A.R.C.S., M.I.Mech.E.

LESSON VIII

PUMPS

Bailing Appliances. Appliances for raising water were probably amongst the earliest machines made by man, and their advance is traceable to the East, where owing to the exigencies of the climate, the practice of irrigation was developed. The first forms were bailing appliances of which the simplest was a vessel lifted by hand, and by adding a rope, water could be drawn from a greater depth. The pulley, windlass, capstan, and treadmill were used as aids in winding the rope. The work of lifting the vessel was eased by suspending it from the longer arm of a counterbalanced lever, as in the Egyptian "shadoof." To save human labour, the power of draught animals, flowing streams, or the wind were utilized, and were arranged to repeat the operation automatically by such means as the bucket wheel, the chain of pots (see Lesson III, Fig. 10) and the chain pump. The scoop wheel and the Archimedian screw were used for low lifts.



Syringe Piston Force Pump Bucket Pump

Fig. 31

Reciprocating Pumps. The idea of a piston reciprocated within a closed cylinder appears to have been first embodied in the syringe. The provision of valves resulted in the force pump, which was known before 200 B.C. The suction pump was in use about the same time, although the reason why the water would not follow the piston to a greater height than about 25 ft. was not understood until the seventeenth century (Fig. 31). The plunger pump, in which the packing is transferred from the piston to the cylinder, was known to the Romans (Fig. 32), and was reintroduced by Sir Samuel Morland about 1675. It is the form used for the highest pressures, and owes its success to the pressure-tightened cup leather packing which is attributed to Henry Maudslay.

The need for elevating large quantities of water for waterworks or mine drainage directed attention to new sources of energy. Savery's engine was applied in 1699, and was followed by Newcomen's atmospheric engine, 1712. For a long period the steam-engine was confined solely to the driving of large pumps, and the work of Watt and his successors led to the highly efficient Cornish engine.

The suction and force pump had been made double-acting at an early period, and by placing the pump barrel in line with the steam cylinder a simpler type resulted. By omitting rotating parts, the piston is free and can act on the water without shock. Valve motion for such pumps was devised by Henry Worthington in 1844, while in his duplex type, introduced about 1850, the valve of one cylinder is controlled by the piston of the other.

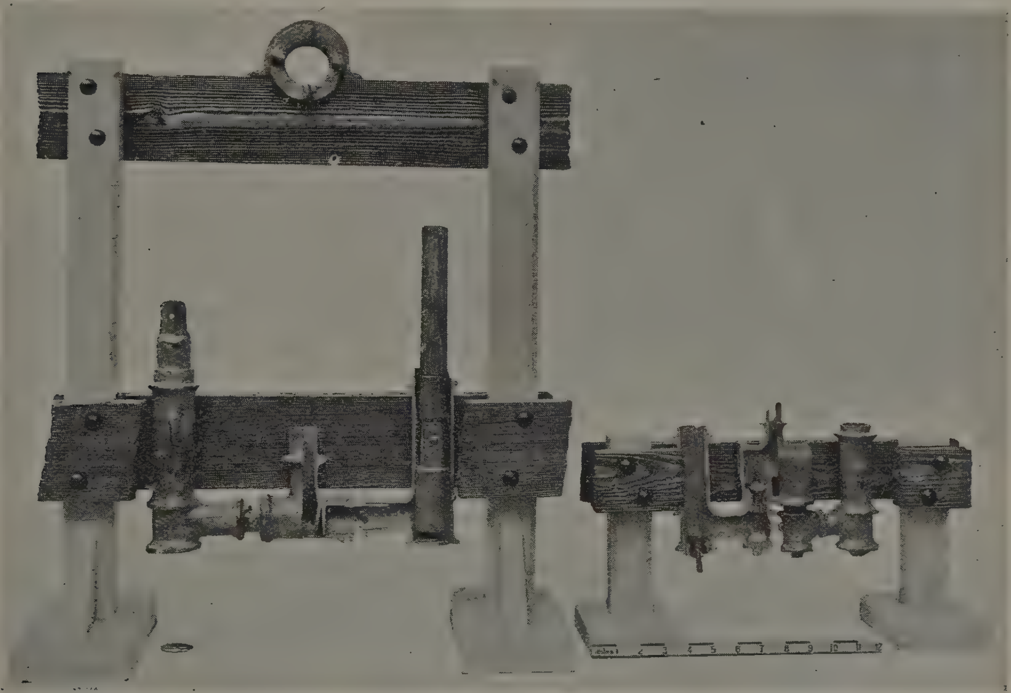
Water-pressure pumps seem to have had their origin in Germany in 1748, and were introduced into England in 1765. The hydraulic ram, which acts by the rise in pressure in a pipe when the flow is suddenly stopped, was first tried by Whitehurst in 1772, and was made self-acting by the Montgolfiers in 1796.

Rotary Pumps. Pumps in which a rotary or oscillating part effects the enlarging or contracting of a pump chamber, were known in the sixteenth century, and they have, in recent times, been much used for low pressures.

In the centrifugal pump, the energy of the prime mover is used to give momentum to the water instead of acting by pressure. Papin, in 1682, devised the Hessian pump with four radial blades arranged in a circular casing, while in 1818 was introduced the Massachusetts pump, which had an eccentric casing. In 1839, Andrews brought out the spiral casing, and, in 1849, J. G. Appold determined the most efficient curve for the blades of the wheel (Fig. 33). About 1850 Prof. J. Thomson suggested the vortex chamber surrounding the wheel. For high lifts the peripheral speed of the wheel must be high, and Gwynne and others employed two or more pumps in series. Increased efficiency has been obtained by forming the passages like those of an outward flow turbine reversed, such as in the Osborne Reynolds pump of 1875.

A form of pump in which the water is displaced by compressed air acting on its surface was used by the Ancients, but it is only in recent years that the system has come into practical use. The idea of exploding a gaseous mixture and allowing the expanding gases to act on the water was proposed in 1868, and was successfully developed by H. A. Humphrey, in 1906, with his internal combustion pump.

Experiments with the air-lift pump were made by Löscher in 1797, but the first practical application was made by Crockford in 1846. The recent extensive adoption of the type dates from 1893, and is largely due to the work of Dr. J. G. Pohle.



By permission of the Director of the Science Museum

FIG. 32. ROMAN PLUNGER PUMPS—1st CENTURY, A.D.

Air Compressors. The earliest air pumps were no doubt skin bags used as bellows, which developed into the common leather bellows, while the Chinese early adopted a double-acting piston pump. When gases are compressed, changes in volume and temperature take place. At the low pressures used for furnace draught the rise of temperature is negligible, but at high pressures it become serious. In the compressors used at the St. Gothard tunnel in 1873, cold water was injected into the cylinder, thus producing the wet compressor. In the dry compressor the cylinder is water-jacketed, while for compressing air to very high pressures, the work is done in stages and cooled between the stages.

Fans. The simple radial bladed fan revolving in a casing is of very early origin, but it has been improved by giving the blades special curved forms, as in the well-known Blackman and Davidson fans. Blowers of the rotary pump type have also been largely used, such as the Beale exhauster of 1857, the Roots blower of 1866, and the more recent rolling drum compressor made by Messrs. Reavell. For dealing with very large quantities of air at moderate pressures, the turbo-compressor was introduced by Parsons in 1901, and this has been succeeded by the multiple centrifugal pump types of Rateau and Sulzer.

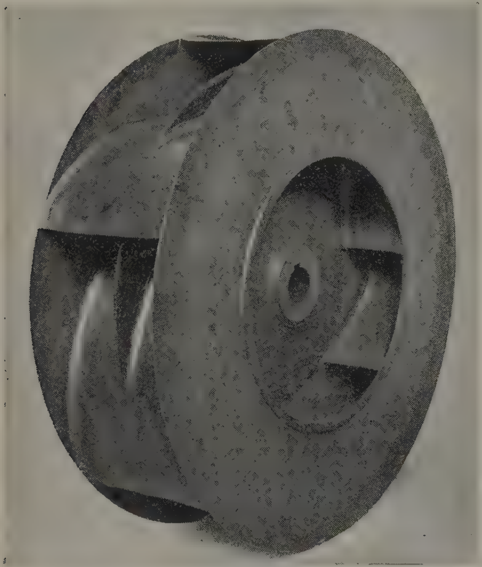


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Fig. 33. Appold Centrifugal Pump, 1855

Injectors. The action of induced currents had been known as early as 1719, but the first practical application seems to have been Trevithick's exhaust steam draught of 1803. Amongst many applications of the principle are Thomson's water jet pump of about 1852; steam jet exhausters and blowers; oil fuel burners, and various forms of ventilator. The most useful application, however, is the injector boiler feeder invented by M. Giffard in 1858 (Fig. 34), and made self-acting by Messrs. Davies & Metcalfe in 1877. The compound form was introduced by Körting in 1873.

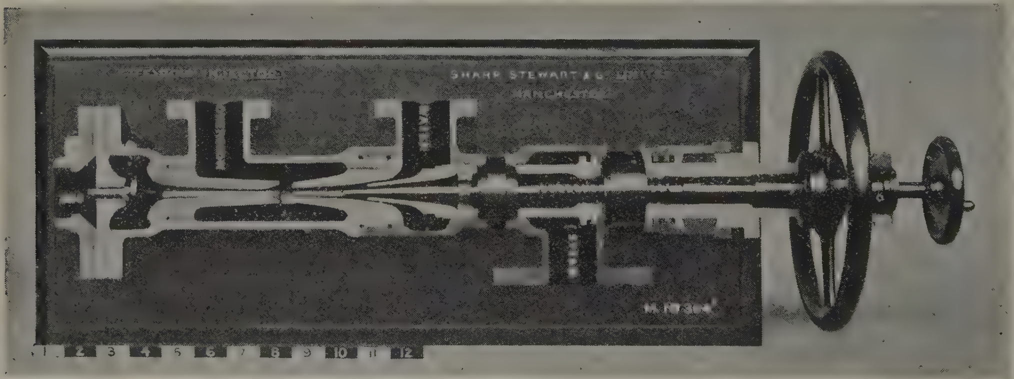


Fig. 34. Giffard's Injector, 1858

METALLURGY FOR ENGINEERS

By James Knowles

LESSON III

CAST IRON

The Calculation of Cupola Mixtures

A glance at the list of purposes of the castings tabulated for the silicon standards in Lesson I, and an understanding of the varied peculiarities of working conditions which the castings are called upon to withstand, during their period of commercial or domestic usefulness, are sufficient to indicate the widely varying nature of the demands the foundry has to satisfy. The power of certain chemical constituents of cast iron to help or hinder in this task has been touched upon. Hence, a definite purpose having been specified, it is necessary to select an iron containing the requisite percentage of the elements desired, together with the smallest possible inclusion of the undesirables. This may be impossible of obtainment from one grade or brand of pig alone, and so a mixture of several must be employed for remelting.

It is, in fact, desirable to use a mixture of different brands. By this means, characteristics due to locality of origin can be counteracted and also, the foundry, being independent of any one source of supply, is saved the embarrassment of a particular brand going off quality.

In these days of accurate chemical analysis and high temperature measurement it is, perhaps, difficult for the lay mind to comprehend how greatly dependent on his eye was the iron or steel maker of former times. Irons were (and still are) classified and graded according to the appearance of the fracture, and temperatures estimated by simply looking at the hot material. Heredity and years of experience combined to bestow a wonderful accuracy of judgment on the individual. In the foundries pig-irons were selected by fracture, mixed by guesswork (inspired by experience) and, by a process of trial and error, first-class castings were produced. A suitable mixture, having been discovered was jealously guarded as a secret, and thus were reputations established and practices evolved which modern science has since consolidated and confirmed.

This eye skill, fortunately, persists to a very large extent to-day, and in many foundries mixing by fracture still obtains. This, however, is to be deplored, since mixing by analysis has so much to recommend it. Briefly, mixing by analysis consists of a selection of irons on their chemical composition and the mathematical calculation of the quantities of the respective components required for a particular charge.

In regard to a particular element, the influence on a mixture of any one component, is proportional to the weight of that component multiplied by the percentage of the element contained in it. Thus, 2 cwt. of an iron containing 2·5 per cent of silicon will give to a mixture five times as much silicon as will 1 cwt. of another containing 1·0 per cent. This is the basis of the "cwt. per cent" method recommended by the Sheffield University.

The whole idea may be expressed by the following simple formulae—

$$\text{cwt.} \times \text{per cent} = \text{cwt. per cent}\,;$$

$$\frac{\text{cwt. per cent}}{\text{cwt.}} = \text{per cent}\,;$$

$$\frac{\text{cwt. per cent}}{\text{per cent}} = \text{cwt.}$$

Space will not permit of more than two examples of its application here, but full details and problems are set out in *General Foundry Practice*, by Macwilliam & Longmuir, Chapter XXIX.

Example 1. A mixture is made up of—

4 cwt.	of pig	containing	2·66	per cent	of	silicon
7 ,,	,,	,,	2·33	,,	,,	,,
9 ,,	,,	,,	2·1	,,	,,	,,

What is the percentage of silicon in the mixture?

	cwt.	×	silicon per cent	=	cwt. per cent silicon
No. 1.	4	×	2·66	=	10·64
No. 2.	7	×	2·33	=	16·31
No. 3.	9	×	2·10	=	18·90
Mixture	20		—		45·85

$$\therefore \text{ Silicon percentage of mixture} = \frac{45{\cdot}86}{20}$$
$$= 2{\cdot}29 \text{ per cent.}$$

Example 2. A 10 cwt. mixture containing 2·75 per cent silicon is required from three irons whose silicon contents are as follows—

No. 1 . 3·10. No. 2 . 2·24. No. 3 . 2·82. Calculate the weights of each.

It is obvious that the bulk of the mixture will be supplied by the iron whose silicon content is nearest that of the mixture, i.e. No. 3. Of this try 6 cwt.

The net effect of the remainder of the mixture must then be to reduce this percentage to 2·75. Since additions of No. 2 will reduce this more than equal additions to No. 1 will increase it, try the result of sharing the remainder of the charge between them.

This is set out as follows—

	cwt.	×	silicon per cent	=	cwt. per cent silicon
No. 3.	6	×	2·82	=	16·92
No. 2.	2	×	2·24	=	4·48
No. 1.	2	×	3·10	=	6·20
Mixture	10	×	—		27·60

$$\therefore \text{ Silicon percentage of mixture} = \frac{27{\cdot}60}{10}$$

$$= 2{\cdot}76 \text{ per cent.}$$

The respective amounts of Nos. 1, 2, and 3 are, therefore, 2 cwt., 2 cwt., and 6 cwt.

The final contents of this charge in regard to the other elements can then be found, if necessary, by applying in each case example 1.

MALLEABLE CAST IRON

1. Carbide of iron, when heated to a good red heat, splits up into carbon and iron. If this temperature is maintained long enough and the subsequent cooling is slow, the reunion of the two elements is prevented, and thus the carbon remains in the free state. But the graphite so precipitated is not the graphite which exists in pig-iron. It will be remembered that this latter variety is in the form of largish flakes, mechanically mixed in the mass, having a general weakening effect on the strength of the iron. In the case under consideration, however, the free carbon is in a very fine state of division, evenly distributed in the metal and, thus, planes of weakness, or cleavage, are not established. This form of graphite is known as amorphous or "annealing carbon."

2. Again, if the heating of the iron is carried out under oxidizing conditions, most of the carbon liberated combines with the oxygen and is thus removed.

3. Carbon exists as a carbide, to any large extent, only in a white iron; in a grey iron it is already largely in the free state, and heat treatment, such as above described, has not the same effect thereon. Grey iron so treated is considerably weakened.

The purpose of these additional notes on the behaviour of the all-important carbide is to furnish the key to the theories underlying the methods of production of malleable cast iron.

Annealed Castings. In many cases the condition of the metal of which grey iron castings are made renders further treatment imperative. For example, small thin castings are liable to a chilling effect at the moment the molten metal comes into contact with the mould. This sudden cooling has the previously noted effect of hardening the skin of the casting, thus militating against easy machining. Furthermore, the very intricacies of the shape of some foundry products will set up serious internal stresses in the metal, resulting in a predisposition to fracture in service. The remedy for both these defects lies in the employment of a corrective heat treatment.

The castings are, therefore, submitted to a slow heating to a good red heat at which they are maintained until a uniform temperature has been attained and are then slowly cooled. The effect of this treatment is to break down the original crystalline formation and substitute a fresh one. Thus are the internal stresses relieved and the hard iron carbide split up into iron and free carbon.

Castings so treated suffer a loss of about 40 per cent of their original tensile strength and are known as "annealed castings" (note 3). They must not be confused with malleable cast iron, hereinafter described.

Malleable Cast Iron. The production of malleable cast iron is achieved by two similar, yet dissimilar methods. Both work on white iron as a basis and may be differentiated as follows—

1. The Reaumur process, whereby the cast iron suffers a loss of carbon contents, whilst

2. Blackheart cast iron results from a treatment calculated to change the combined carbon in the white iron into annealing carbon.

The Reaumur Process. Though first outlined by the French scientist whose name it bears, as long ago as 1722, this method was patented by Lucas, of Sheffield, in 1804, and is the typical English process.

The original composition of the metal, usually white or mottled Cumberland hematite, is very important since it must leave the mould as a hard white iron, perfectly free from graphite. Therefore variations in the sizes and rates of cooling must be compensated by the percentage of silicon. (See Table of Silicon Standards on page 620.)

The castings, which have been fettled to remove adhering sand, etc., are packed in cast-iron pots along with an oxidizing material, such as red hematite ore, mill scale, etc. The packing must be done in such a way as completely to isolate the castings from each other, and to prevent distortion due to heat. The pots are closed by cast-iron lids, all air spaces being made up with clay, and placed in the annealing oven. This usually consists of a four-sided chamber with fire grates, or gas inlets, at the corners and below the general floor level. The flames thus enter the annealing furnace at, or about, floor level, play round the pots, and are finally drawn out by means of a flue running down the centre of the roof. The temperature is slowly raised during two or three days until about 850 to 880° C. has been reached, which is maintained for a period of three to four days, when slow cooling, over a further period of about four days, is begun.

At the heat thus attained the free carbon, liberated from the iron carbide, is enabled to unite with the oxygen of the packing to form carbon monoxide (Note 2). As a result of this reaction a loss of about 97 per cent of the original carbon may be sustained.

Blackheart Castings. These are so called on account of the appearance of the fracture, which presents a velvety black "heart," or core, surrounded by silky grey edges. The practical working of this process, extensively used in America, is almost identical with that of the foregoing, with the important difference that the packing materials used are non-oxidizing, e.g. sand, etc. Temperatures, also, run somewhat lower, 770 to 840° C. representing average practice. The only purpose of the packing is to prevent the scaling of the castings due to heat. The theory of the treatment is, of course, that indicated in Note 1.

For some reason, sulphur plays a very destructive part, if present to any great amount, in the products of this practice and, hence, American blackheart castings are kept as free as possible from this element, whilst the manganese content generally rules higher than in the Reaumur process. British white irons are usually too high in sulphur to render them a suitable basis for blackheart malleable cast iron.

The Results of the Heat Treatment on the Mechanical Properties. The results of the heat treatment on the mechanical properties of the metal are best demonstrated by the following comparative table, which gives approximate figures—

	Maximum Stress	Elongation on 2 in.	Reduction of Area
White cast iron	11 tons sq. in.	—	—
Reaumur malleable cast iron	25 ,, ,,	5 per cent	6 per cent
Blackheart	20 ,, ,,	10 ,,	10 ,,
British Admiralty specification	18 ,, ,,	4·5 ,,	—

PIONEERS OF ENGINEERING

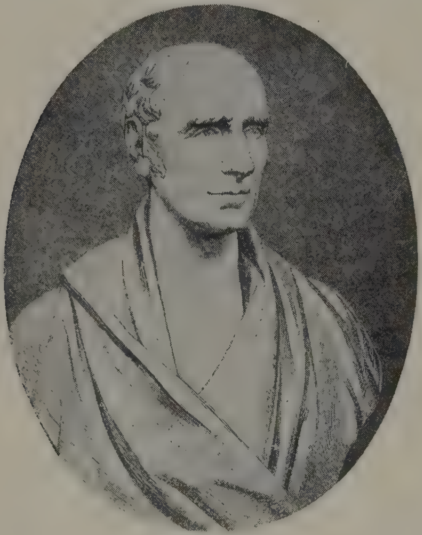
By J. F. Corrigan, M.Sc., A.I.C.

LESSON XV

GAS LIGHTING INVENTED

WILLIAM MURDOCK (1754–1839)

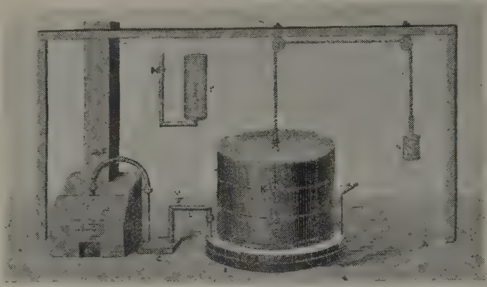
The commercial and industrial application of gas lighting cannot definitely be ascribed to any one person. Nevertheless, it is to William Murdock that the honour must be credited of contributing more practical and initial benefit to the subject than any other one person.



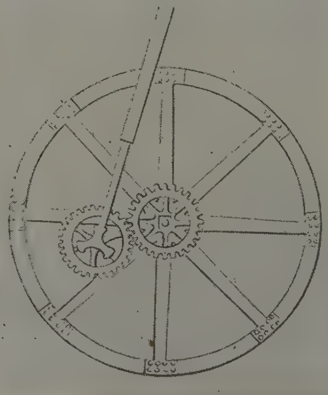
William Murdock

Murdock was by profession a mechanical engineer. He was, in fact, employed by Watt to superintend the building of his engines, and later on Murdock became Watt's most trusted associate and co-worker. Murdock's father was a millwright at Bellow Mill, near Old Cumnock, in Ayrshire, and it was in that locality that his famous son was born in 1754. Young William was brought up in his father's trade. But the routine occupation of the father did not fit in with the ideas of the son. Consequently, William, arrived at early manhood, proceeded to Birmingham, and obtained employment with the firm of Boulton and Watt. He was first employed with the concern as an ordinary workman, after which he travelled up and down the country, superintending, as we have noted above, the erection of motive engines.

Murdock's first practical experiments in the subject of gas lighting were made at a time when he was stationed at Redruth, in Cornwall. He heated various substances in an iron retort, and conducted the inflammable gases which were evolved through copper tubes to suitable burners. In this manner, he managed to light a whole room with the gases evolved by the distillation of coal. In 1797, on his return to Scotland, he lighted the mill premises at Old



An Early System of Gas Manufacture
(From an Old Print)



The "Sun and Planet" Motion

Cumnock with coal gas. In the following year, the works of Boulton and Watt at Birmingham were similarly treated.

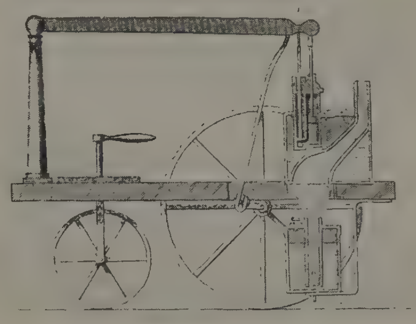
That is about as far as Murdock went in the purely practical application of coal gas lighting. Other people took the subject up after witnessing the success of his early efforts, and shortly afterwards one or two of London's principal thoroughfares were lighted with the gas.

The introduction of coal gas was not altogether popular. People saw in it a dangerous, poisonous, and a (considered) impracticable method of obtaining illumination, and no doubt there were many vested interests against its commercial adoption. One verse of the period runs—

> Must Britons be condemned for ever to wallow
> In filthy soot, noxious smoke, train oil, and tallow
> And their poisonous fumes for ever to swallow ?
> For with sparky soots, snuffs, and vapours, men have constant strife ;
> Those who are not burned to death, are smothered during life.

Nevertheless, gas lighting very quickly triumphed over its adversaries, and within a score of years after Murdock's early trials, the new method of lighting was being adopted all over the country.

In the realm of steam and mechanical engineering, Murdock is celebrated in a no less manner than he is in the sphere of gas lighting. Many of Watt's detail improvements in his engines were, in reality, due to Murdock. It is to Murdock that the origin of the "Sun and Planet" circular motion as applied to connecting rods is due. Murdock also invented a locomotive which was, to some extent, a practical success.



Murdock's Locomotive

In all, therefore, William Murdock lived a full life. He died in 1839 at the advanced age of 84,

ENGINEERING SPECIFICATIONS

By Arthur P. Haslam, M.I.E.E.

LESSON VI

EXAMPLES OF ACTUAL SPECIFICATIONS

It will help us in our understanding of the preceding chapters if we see how the principles we have considered are applied in actual practice. The best way to do this is to examine a few specifications of standard engineering productions.

Lancashire Boiler. Taking, in the case of Lancashire boilers, the "John Thompson" make as typical, we find—

Delivery. Purchasers must take responsibility that all roadways and bridges to be passed over are suitable and safe to carry the weight nvolved.

Erection. Purchasers must find unskilled abour and timber packing.

Inspection. Tests to be carried out at maker's works before delivery. Special material tests by any outside authority to be made at purchaser's expense unless specifically included in estimate.

Performance. A performance guarantee shall contain a bonus and penalty clause, and a penalty shall be as liquidated damages. In the event of penalties being claimed, reasonable time and opportunity for alterations shall be allowed.

Delivery. No liability to be incurred unless date in contract is accompanied by a bonus and penalty clause, in which case penalty shall be as liquidated damages. If special causes outside manufacturer's control cause delay, reasonable extension shall be granted.

Maintenance. Manufacturer's liability to be strictly limited to making good any defects arising solely from faulty workmanship or material, and it does not cover defects due to improper use. No liability is accepted for boilers going abroad.

Arbitration. In case of dispute, an arbitrator is—failing agreement—to be appointed by the President of the Institution of Mechanical Engineers. Neither party is to be represented at any inquiry by solicitor or counsel.

Material. This important point is specified as follows—

The boiler shall be made entirely of steel, which shall be subject to the following tests—

Strips cut from the shell plates to be pulled asunder in a testing machine, and shall afford a tenacity of not less than 28 tons, and not more than 32 tons, with an elongation of 20 to 25 per cent in a length of 10 in. Other strips cut from the plates shall be heated to a cherry red, and cooled in water of 80° F., and then bent without fracture with an inner radius of not more than four times its own thickness.

Strips from the fire-box, and flues for the plain flanged portions shall afford a tenacity of not less than 24 tons and not more than 28 tons per sq. in., with an elongation of 20 to 25 per cent in a length of 10 in.

Each plate to be branded and numbered in accordance with the test certificate of the material.

Rivets to be of best selected rivet steel, the tensile test of which shall be 25 to 30 tons per sq. in., with an elongation of not less than 25 per cent in a length of 10 in.

Design. The "John Thompson" boiler has the front and back end plates of a convex shape. This, together with the arrangement of the shell plates, the manner in which the rivet holes are placed, the construction of the flues, are all described in the specification, which also mentions the general method of manufacture. The rest of the specification describes and specifies in detail the mountings and accessories which are included in the contract.

Water Tube Boilers. The same firm manufacture several types of water tube boilers. They have the vertical straight tube, the vertical bent tube, and the stepped header horizontal tube types, and specify one or the other pattern to suit particular cases.

So far as the quality of the steel plates is concerned, the specifications for all these types state that—

> All the plates to be of best quality Siemen's Martin mild steel, with a tensile strength of not less than 24 tons or more than 28 tons per sq. in. for all pressed plates, and a tensile of not less than 28 tons or more than 32 tons per sq. in. for the shell plates, with an elongation of not less than 20 per cent in a length of 8 in.

As regards riveting they state—

> The riveting, whenever practicable, to be performed by hydraulic pressure and done slowly so that each rivet may be cooled down to a black heat before the pressure is removed. All rivets to have substantial heads, and completely fill the rivet holes, and after riveting the seams to be carefully caulked by pneumatic hammers.

Another well-known maker of water tube boilers and of steam-raising accessories is Messrs. Babcock & Wilcox, Ltd. In their marine type boiler they treat the boiler with its superheater and superposed economizer as one unit, and specify as its capacity—

> The unit to be capable of evaporating — lb. of water per hour from a feed temperature of —° F. entering economizer into steam of —lb. per sq. in. pressure at the superheater outlet, the superheater to be capable of superheating the steam —° F. to —° F. to give a final temperature of —° F.

This information coupled with the total heating surface and total grate surface of the boiler, and the total heating surface of the super-heater and economizer enables the normal capacity of the boiler under working conditions to be determined.

The material used is thus specified—

> The headers to be solid drawn from open hearth steel billets of the best quality, having a tensile strength of 26–30 tons per sq. in. and an elongation of not less than 25 per cent in 8 in. The plates for the steam and water drums to be of the highest quality open hearth mild steel, the shell plates to have a tensile strength of 28–32 tons per sq. in. with an elongation of not less than 20 per cent in 8 in. The plates for drum ends to have a tensile strength of 26–30 tons per sq. in. with an elongation of not less than 23 per cent in 8 in.

The responsibility for maintenance clause is thus worded—

> All materials and workmanship to be first class in every particular, but supplier's liability is limited to the replacement without charge to the purchasers, f.o.r. at supplier's works of any part of the boiler or boilers, the subject of this specification which under proper and normal conditions of working within one year from date of delivery requires replacement in consequence of any original defect in workmanship or material.

The responsibility of the suppliers for delivery is thus expressed—

> To commence in —— weeks unless prevented by circumstances beyond our control, such as, fires, floods, strikes, lock-outs, riots, war conditions, accidents in our works or in those furnishing us with material, and delays of transportation or carriers.

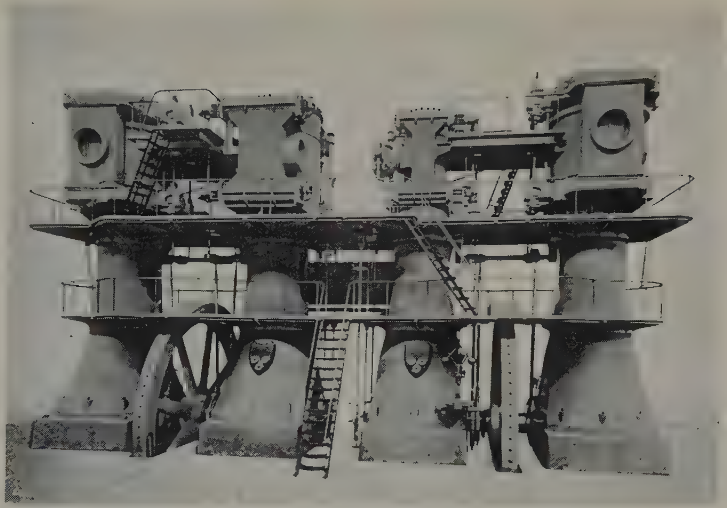


FIG. 28. STEAM BLOWING ENGINE. FOUR-CRANK COMPOUND TYPE

(Messrs. Richardsons, Westgarth & Co.)

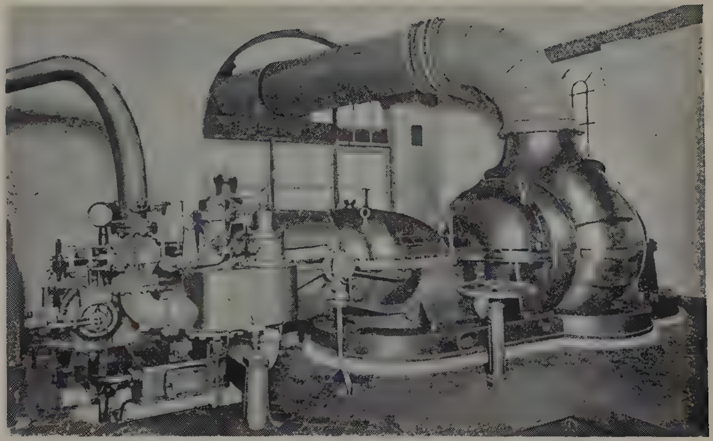


FIG. 30. DOUBLE-FLOW BLAST FURNACE TURBO-BLOWER, DRIVEN BY PARSONS STEAM TURBINE

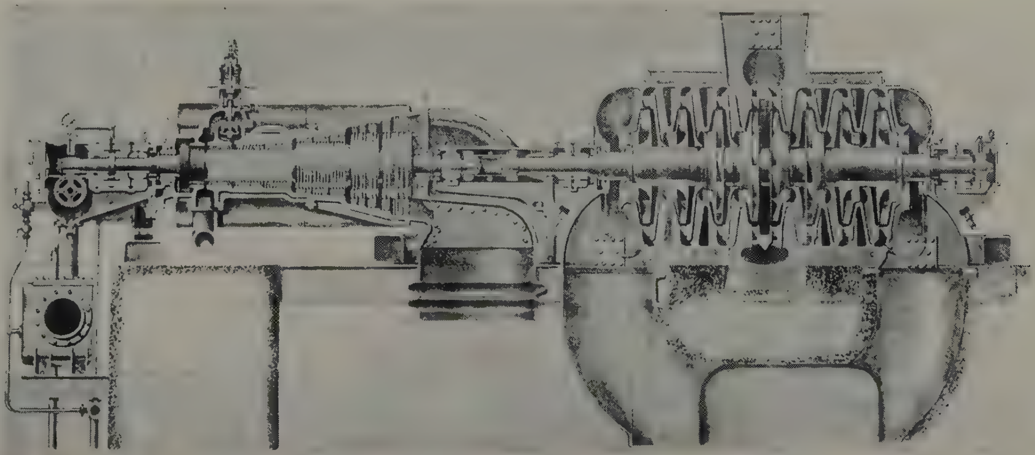


FIG. 29. SECTION THROUGH PARSONS STEAM TURBINE-DRIVEN TURBO-BLOWER

BLAST FURNACE PRACTICE

By PERCY C. R. KINGSCOTT, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (LOND.)

LESSON VII

BLOWING PLANT

BOILERS

BEFORE the economical use of waste gas became of paramount importance in the steel industry, it was commonly the practice to burn blast furnace gas wantonly in boilers of the Lancashire type provided with inefficient burners and combustion chambers. The gas was kept burning by the use of coal, and when the gases available were reduced in quantity, or when the furnaces were stiff, steam-raising was extremely difficult, even by using coal in quantity, at precisely the moment when the maximum effort was required of the steam-blowing engine. Of late years, burners and combustion chambers have received attention, as with the hot blast stove, and similar apparatus has been adopted. In such installations, water-tube boilers, such as the Babcock and Wilcox, the Stirling, the Woodeson, and the Nesdrum have replaced the plain cylindrical types referred to above. A spare unit should be allowed for use during repairs or cleaning. Fig. 26 shows a battery of "Nesdrum" water-tube boilers, suitable for using blast furnace gas, in course of erection, the photograph being reproduced by permission of Messrs. Richardsons, Westgarth & Co., of Middlesbrough, makers of this boiler. This type of boiler, in conjunction with efficient water softeners and power tube cleaners, has increased the efficiency and decreased the costs of steam raising.

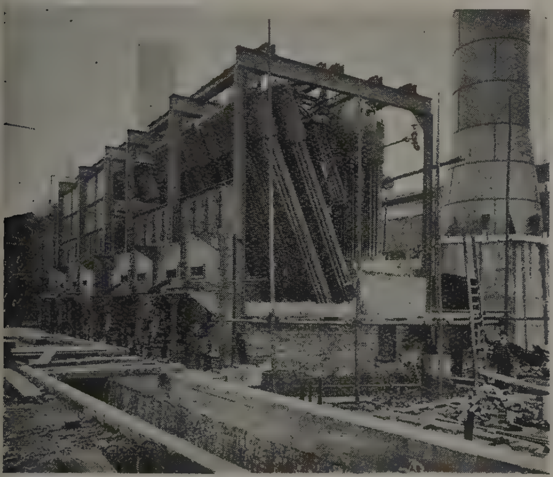


FIG. 26. "NESDRUM" WATER-TUBE BOILERS

BLOWERS

Early forms of modern blowers were driven by reciprocating steam-engines usually of the vertical type. There are still isolated cases of water-driven blowers, but these are of small importance. The chief improvements made

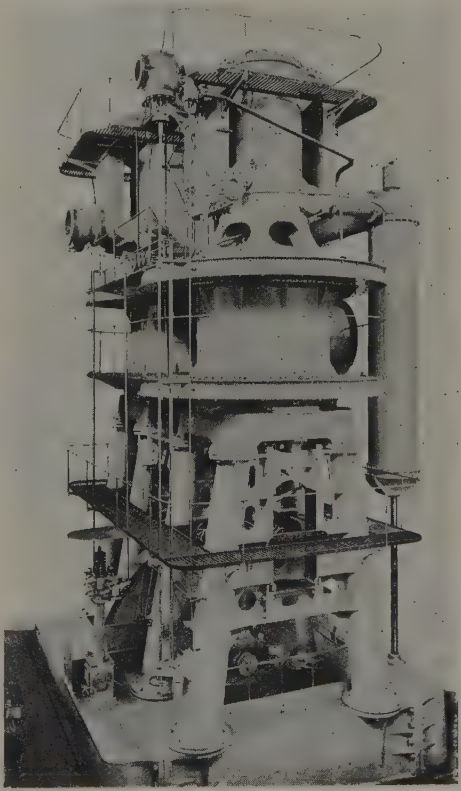


FIG. 27. STEAM BLOWING ENGINE. COMPOUND TANDEM TYPE

(Messrs. Richardsons, Westgarth & Co.)

were due to the increased demand for higher pressures and larger volumes of air, and concerned chiefly the valve gear. The latter was originally automatic, but was later changed to one positive in working. Fig. 27 shows a vertical tandem compound blower with air cylinders beneath the steam cylinders, which are fitted with Corliss gear.

Fig. 28 shows a set of quarter-crank blowing engines, consisting of one high pressure engine driving air cylinder through the crank, and one low pressure engine driving a similar cylinder through its crank. These engines can be run together or separately, the steam cylinders are fitted with Corliss gear and the blowing cylinders with the Southwark sliding air valves. These illustrations are reproduced by permission of Messrs. Richardson, Westgarth & Co., Middlesbrough.

The **turbo-blower** is now largely used in steam-driven plant, considerable advance having been made of late years in the size, suitability, and efficiency of both the steam turbine and the turbo-blower.

Fig. 29 shows a longitudinal section of a Parsons turbo-blower, capable of blowing 40,000 cub. ft. of free air per minute at a normal pressure of 14 lb. to the sq. in., and to give a maximum output of 36,000 cub. ft. per minute at a pressure of 23½ lb. gauge.

Fig. 30 shows the station view of a turbo-blower whose normal duty is 36,000 cub. ft. per minute of free air at 10 lb. gauge.

These figures are reproduced by permission of Messrs. C. A. Parsons & Co., Ltd., Newcastle-upon-Tyne.

Gas Engine Blowers. Since the installation in 1895, by Thwaites, of a gas engine operating on blast furnace gas, considerable advance has been made in this practice, particularly as the result of the elimination, by European gas engine experts, of mechanical and technical difficulties, and the introduction of efficient gas cleaners. Valve design in the blowers has returned to the automatic type.

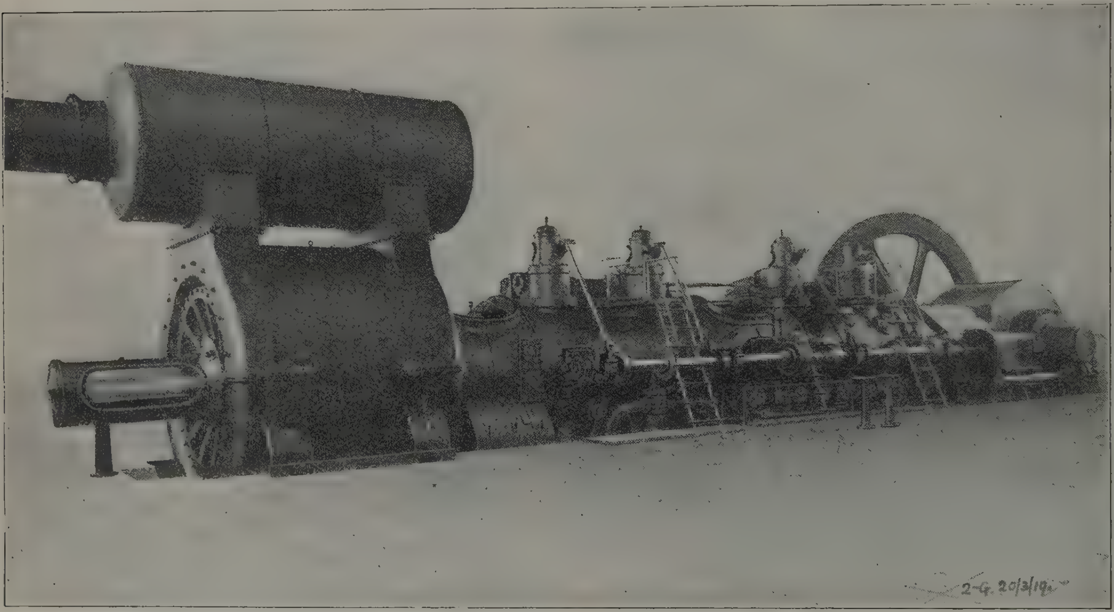


FIG. 31. GAS BLOWING ENGINE
(Messrs. Galloways)

Fig. 31 shows a gas blowing engine built by Messrs. Galloways, Ltd., Manchester, with whose permission the photograph is reproduced here. This engine has a capacity of 35,000 cub. ft. air at 10 lb. per minute at 85 r.p.m. Cylinders, 1,150 mm., stroke, 1,300 mm. The blowing cylinder shown in Fig. 32 is connected to the rear engine cylinder by a distance piece. The air cylinder barrel is spigoted into two air

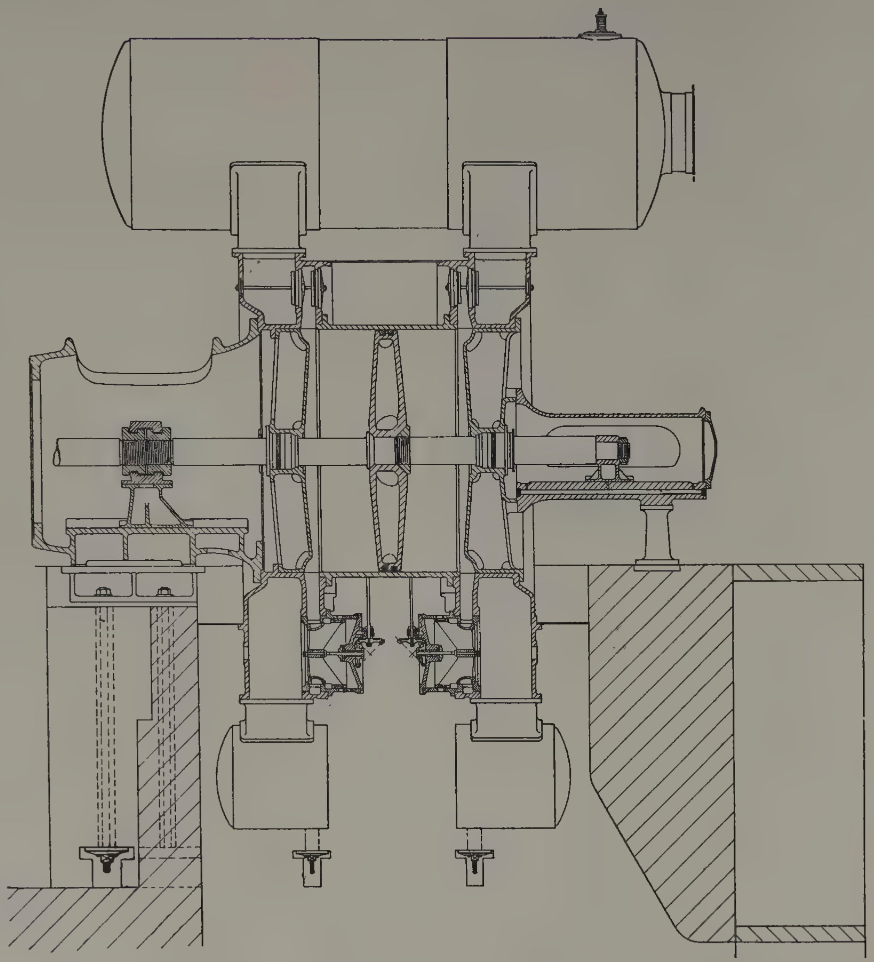


Fig. 32. Section of Blowing Cylinder

(Messrs. Galloways)

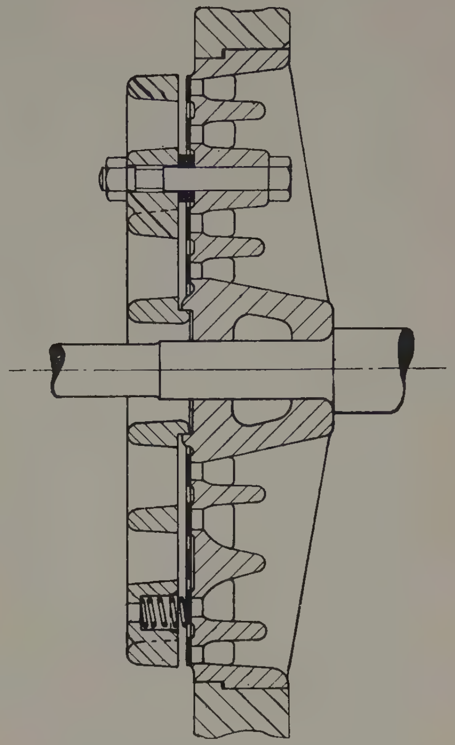


Fig. 33. Blowing Cylinder Air Valve

(Messrs. Galloways)

heads containing the suction and delivery valves. A large air belt round the air cylinder barrel is formed by a cylindrical steel jacket fitted between the air heads. The suction valves are in the inner sides of the air heads next to the air belt. Air enters the latter from a chamber in the foundations. The blowing cylinder air valves are of patented suspended frictionless type, made of special steel in the form of discs. (See Figs. 33 and 34.) These have concentric annular slots and are provided with three annular segments which act as light springs, and by which the valve is suspended between the valve seat and valve guard, which are both multiple ported. Small spiral springs in recesses in the guard ensure the efficient seating of the valve.

Gas must be cooled to about 20° C., and cleaned to contain the minimum amount of dust. Hence, the latter must be undertaken in a Halberg-Beth apparatus, or in a secondary wet washer, such as the Theisen, Fowler and Medley, or Schwartz Bayer, using a primary washer, such as the Brassert static plant.

General opinion avers that gas engines are more suitable for blowing engines than for electrical prime movers. It is probably advisable in a steel works to use gas engine blowers provided with exhaust heat boilers, and to use a combination of turbo-generators and gas engine alternators for power purposes. Surplus gas, after providing for the stoves, can then be used to advantage in metallurgical processes.

Where the surplus gases cannot be used, e.g. in a furnace plant unattached to a steel works, it is probably advisable to use high pressure reciprocating engines or turbo-blowers.

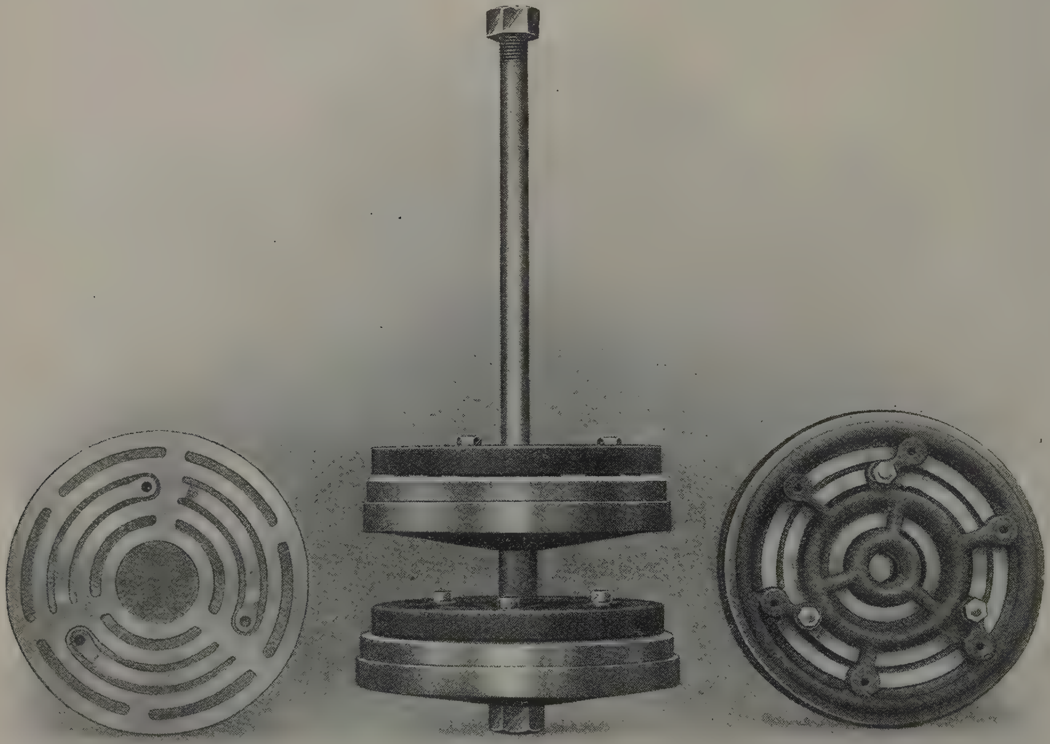


Fig. 34. Blowing Cylinder Air Valve
(Messrs. Galloways)

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
Lecturer in Mathematics, Battersea Polytechnic

LESSON XVI

TRIGONOMETRY—(*contd.*)

Sine and Cosine (*contd.*). In the last lesson we had just introduced sine and cosine, and worked a few straightforward examples in their use. We proceed to give further examples.

Just as with the tangent of an angle, sines and cosines may be used to find angles, and we give the formulae derived from (*C*) and (*D*) of Lesson XV for this. Dividing by *h*, we have

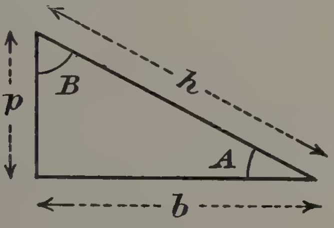


Fig. 53

$$\sin A = \frac{p}{h} \quad . \quad . \quad . \quad (E)$$

$$\cos A = \frac{b}{h} \quad . \quad . \quad . \quad (F)$$

where the letters stand for the sides and angles of Fig. 53, reproduced from the preceding lesson.

Example 3. The gradient of a hill is 1 in $6\frac{1}{2}$; what is the angle of slope in degrees?

The data are represented in Fig. 61, a rise of 1 occurring in a distance 6·5 measured along the slope. We have, then,

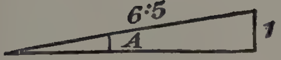


Fig. 61

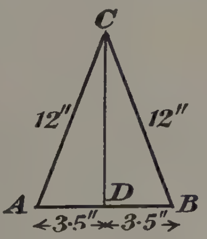


Fig. 62

$$\sin A = \frac{1}{6{\cdot}5} = 0{\cdot}1538$$

whence, $A = \underline{8^\circ\ 51'}$ (from table).

4. The equal sides of an isosceles triangle are 1 ft. long, and the base is 7 in. long. Find the angles of the triangle.

A sketch is given in Fig. 62, the triangle being divided into two right-angled triangles by a perpendicular to the base, and all dimensions marked in inches. Using the triangle *ACD*, in which the hypotenuse is 12 in. and the base 3·5 in., we have

$$\cos A = \frac{3{\cdot}5}{12} = 0{\cdot}2917$$

whence, $A = \underline{73^\circ\ 2'}$ (from table)

consequently $B = A = \underline{73^\circ\ 2'}$

and $C = 180^\circ - 2 \times 73^\circ\ 2'$

$= \underline{33^\circ\ 56'}$

5. In the roof truss illustrated in Fig. 63, the span *AB* is 24 ft., and the angles *A* and *B* each 25°. Find the length of the sloping members *AC* and *BC*.

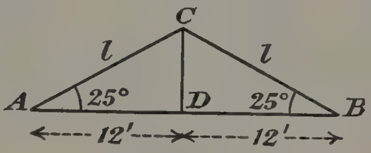


Fig. 63

The isosceles triangle is again divided into two equal right-angled triangles, and, considering the triangle *ACD*, we know the base, $AD = 12$ ft., and the angle $A = 25^\circ$; to find the hypotenuse, *l*, we use (*D*), giving

$$12 = l \cos 25^\circ$$

whence $l = \dfrac{12}{\cos 25^\circ} = \dfrac{12}{0{\cdot}9063}$

$= \underline{13{\cdot}24 \text{ ft.}}$

Remarks on tan, sin, and cos. It should be evident that, since the hypotenuse of a right-angled triangle is greater than each of the other sides, the sine and cosine of any angle can never be greater than 1. For a very small angle, the base and hypotenuse are very nearly equal, while the perpendicular is small, so the cosine is very nearly 1, and the sine is small. In the limit,

$$\sin 0^\circ = 0, \text{ and } \cos 0^\circ = 1.$$

Further, when the angle is nearly 90°, the perpendicular is nearly equal to the hypotenuse, and the base is small. The sine is then nearly 1, and the cosine small. In the limit,

$$\sin 90^\circ = 1, \text{ and } \cos 90^\circ = 0.$$

Thus, as the angle increases from 0° to 90°, the sine *increases* from 0 to 1; while the cosine *decreases* from 1 to 0, as the angle increases from 0° to 90°. On account of the decrease of the cosine, care is needed in using the difference columns of the tables—differences must be *subtracted* where in *log.*, *sin*, and *tan* tables they must be added, and vice versa. The values of the tangent are not restricted in the same way as those of the sine and cosine. The tangent can have *any* value. At 45°, the value of the tangent is 1; for angles less than 45° it is less than 1, and for angles greater than 45°, it is greater than 1. In the neighbourhood of 90°, the tangent increases without limit,

$$\tan 90^\circ = \infty \text{ and } \tan 0^\circ = 0.$$

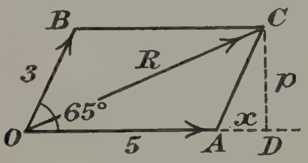


FIG. 64

Complementary Angles. In any right-angled triangle, there are two other angles, and these are complementary, i.e. their sum is 90°. The same sides are used to build up the sin, cos, and tan of each. Thus, in Fig. 53,

$$\sin A = \frac{p}{h} = \cos B, \therefore \sin A = \cos (90^\circ - A)$$

$$\cos A = \frac{b}{h} = \sin B, \therefore \cos A = \sin (90^\circ - A)$$

$$\tan A = \frac{p}{b}, \quad \tan B = \frac{b}{p}$$

$$\therefore \tan (90^\circ - A) = \frac{1}{\tan A}$$

These relations should be known.

Further Examples on Sin, Cos, and Tan. To proceed in showing the uses of these trigonometrical ratios, as they are called (for reasons which (*B*), (*E*), and (*F*) should make clear), we will show how they are applied to triangles not right-angled—though this has been done, to some extent, in dealing with the tangent.

EXAMPLE 6. Find the resultant of two forces of 5 lb. and 3 lb., inclined at 65°, and the direction of the resultant.

The forces are represented by *OA* and *OB* in Fig. 64, where *OC* represents the resultant, *R*. Drawing *CD* perpendicular to *OA* produced, the triangle *CAD* is right-angled, and in it, $AC = 3$ and $\angle CAD = 65^\circ$. So, calling *AD* x, and *CD* p, we have

$$p = 3 \sin 65^\circ = 3 \times 0{\cdot}9063 = 2{\cdot}7189,$$

$$x = 3 \cos 65^\circ = 3 \times 0{\cdot}4226 = 1{\cdot}2678.$$

Then, $$OD = 5 + x = 6{\cdot}2678,$$

and thus $$\tan \angle COD = \frac{CD}{OD} = \frac{2{\cdot}7189}{6{\cdot}2678} = 0{\cdot}4338,$$

$$\therefore \angle COD = 23^\circ\, 27'$$

Also, $$OD = R \cos \angle COD$$

i.e. $$6{\cdot}2678 = R \cos 23^\circ\, 27'$$

$$= R \times 0{\cdot}9174$$

$$R = \frac{6{\cdot}2678}{0{\cdot}9174} = \underline{6{\cdot}832 \text{ lb.}}$$

(It would have been possible to use the " square rule to find *R*, after finding p and *OD*, but the method used above is preferable, as it involves less calculation.)

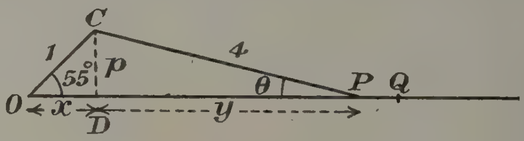


FIG. 65

7. The crank of an engine is 1 ft. long, and the connecting rod is 4 ft. long. Find the distance of the piston from the end of its stroke when the crank has turned 55° from the inner dead centre.

In the diagram (Fig. 65) *OC* is the crank, *CP* the connecting rod, and *OPQ* the line of stroke. *Q* is the end of the stroke, so that $OQ = 1 + 4 = 5$ ft. We shall use the triangle *OCP* to find *OP*, and then obtain *PQ*, the required distance, by subtraction. Draw *CD* perpendicular to OP, call OD x and DP y. Then, in the triangle OCD,

$$p = 1 \sin 55^\circ = 0{\cdot}8192 \text{ ft.}$$
$$x = 1 \cos 55^\circ = 0{\cdot}5736 \text{ ft.}$$

Calling $\underline{/CPD}$ θ, we have, in the triangle CPD,

$$p = 4 \sin \theta$$
$$\sin \theta = \frac{p}{4} = \frac{0{\cdot}8192}{4} = 0{\cdot}2048$$

whence $\theta = 11^\circ 49'$

Now, also,

$$y = 4 \cos \theta = 4 \times 0{\cdot}9788 = 3{\cdot}9152 \text{ ft.}$$
$$OP = x + y = 0{\cdot}5736 + 3{\cdot}9152$$
$$= 4{\cdot}4888 \text{ ft.}$$

so $PQ = 5 - OP = 5 - 4{\cdot}4888 = \underline{0{\cdot}5112 \text{ ft.}}$

EXERCISE No. 26

1. A hill slopes at an angle of 15°. How far does one rise vertically in going 220 yd. along the slope?
2. The gradient of a hill is 1 in $4\frac{1}{4}$. At what angle does it slope?
3. A ship steams 35 miles in a direction 18° W. of N. Find its westing and northing.
4. The sides of a roof are each 12 ft. wide and slope at 35° to the horizontal. Find the height of the ridge above the eaves, and the width of the building.
5. The sides of a rhombus are 4 in. long, and the longer diagonal is 7 in. long. Find the angles, and the length of the other diagonal.
6. Find the magnitudes and directions of the resultants of the following pairs of forces. (*a*) 3 and 5 lb., at 50°. (*b*) 10 and 4 tons, at 120°.
7. An aeroplane is being steered due N. at 100 m.p.h. (relative to the air) in a N.E. wind blowing at 35 m.p.h. At what speed, and in what direction, does it fly as judged from the ground?
8. If it is desired to fly due N. under the conditions of the last example, in what direction must the aeroplane be steered, and what will be the speed relative to the ground?
9. The crank of an engine is 9 in. long, and the connecting rod 45 in. Find the distance the piston has moved from the end of its stroke for every 15° of angle turned through by the crank, up to half a revolution, and plot the displacement-angle graph.
10. Find the angles of the triangle in Ex. 23 (1*c*).

Other Trigonometrical Ratios. Referring to Fig. 53, and formulae (B), (E), and (F), we see that the three ratios, tan, sin, and cos, include all the possible pairs of sides, and thus are sufficient to deal with any case. It is, however, convenient to use their reciprocals sometimes, and these too are given names.

The reciprocal of the tangent is called the **cotangent** (written *cot*, for short), i.e.

$$\cot A = \frac{b}{p} = \frac{1}{\tan A} \quad . \quad . \quad (G)$$

Remembering the relations for complementary angles, we also have

$$\cot A = \tan (90^\circ - A)$$

The reciprocal of the cosine is called the **secant** (written *sec* for short), i.e.

$$\sec A = \frac{h}{b} = \frac{1}{\cos A} \quad . \quad . \quad (H)$$

The reciprocal of the sine is called the **cosecant** (written *cosec* for short), i.e.

$$\operatorname{cosec} A = \frac{h}{p} = \frac{1}{\sin A} \quad . \quad . \quad (I)$$

Remembering the relations between *sin* and *cos* of complementary angles, then by (H) and (I),

$$\sec A = \operatorname{cosec} (90^\circ - A), \text{ and}$$
$$\operatorname{cosec} A = \sec (90^\circ - A).$$

Thus, any ratio is the co-ratio of the complementary angle. Some collections of tables contain only sines, cosines, and tangents; others contain only sines, tangents, and secants. By putting in the complementary angles as well, all six ratios may be found from tables of the latter type. Tables containing all six are certainly more convenient, however. By means of the six, any side can be obtained in terms of any other side, and either of the non-right angles, by multiplication.

Relations Between the Ratios. The "square" rule for right-angled triangles gives us a set of relations between the six ratios which are very important, and should be remembered. Still referring to Fig. 53, we know that

$$h^2 = b^2 + p^2.$$

Dividing through by h^2,

$$1 = \frac{b^2}{h^2} + \frac{p^2}{h^2}$$

or,

$$1 = \cos^2 A + \sin^2 A,$$

where $\cos^2 A$ means the square of the cosine of A, and so on. This relation is really, as is evident from its derivation, the trigonometrical expression of the theorem of Pythagoras.

Again, dividing by b^2

$$\frac{h^2}{b^2} = 1 = \frac{p^2}{b^2}$$

i.e. $\sec^2 A = 1 + \tan^2 A$, or $\tan^2 A = \sec^2 A - 1$.

Again, dividing by p^2,

$$\frac{h^2}{p^2} = \frac{b^2}{p^2} + 1$$

i.e. $\operatorname{cosec}^2 A = \cot^2 A + 1$, or $\cot^2 A = \operatorname{cosec}^2 A - 1$

There are also two more important relations, for

$$\frac{\sin A}{\cos A} = \frac{p/h}{b/h} = \frac{p}{b} = \tan A$$

and, similarly,

$$\frac{\cos A}{\sin A} = \cot A.$$

By the use of these formulae, or by the triangle itself (which is better in numerical work), we can, given any one of the ratios, find the other five.

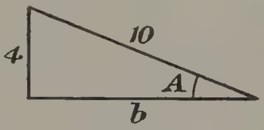


Fig. 66

1. Given $\sin A = 0{\cdot}4$, find all the other ratios. To do this, we may imagine a triangle in which $\frac{p}{h} = 0{\cdot}4$, and this can be done in many ways. Perhaps the most obvious is to have $p = 4$ and $h = 10$, as in Fig. 66. Then, to find b,

$$b^2 = 10^2 - 4^2 = 84$$

$$b = \sqrt{84} = 9{\cdot}165.$$

Then
$$\cos A = \frac{b}{h} = \frac{9{\cdot}165}{10} = \underline{0{\cdot}9165}$$

$$\tan A = \frac{p}{b} = \frac{4}{9{\cdot}165} = \underline{0{\cdot}4364}$$

$$\sec A = \frac{h}{b} = \frac{10}{9{\cdot}165} = \underline{1{\cdot}0910}$$

$$\operatorname{cosec} A = \frac{h}{p} = \frac{10}{4} = \underline{2{\cdot}5}$$

$$\cot A = \frac{b}{p} = \frac{9{\cdot}165}{4} = \underline{2{\cdot}2912}$$

Whichever of the ratios we have to start with, we can always find a triangle in which we know two sides, and by calculating the third, find all six ratios. As an example of the use of the formulae, without appeal to a figure, we will work.

2. Given $\tan A = 1{\cdot}5$, find the other five ratios.

Referring to the formulae, we see that the one which enables us to calculate another ratio from the tangent

is $\quad \sec^2 A = \tan^2 A + 1 = 2{\cdot}25 + 1 = 3{\cdot}25$

so, $\quad \sec A = \sqrt{3{\cdot}25} = \underline{1{\cdot}8028}$

Then, $\quad \cos A = \frac{1}{\sec A} = \frac{1}{1{\cdot}8028} = \underline{0{\cdot}5548}$

$$\sin^2 A = 1 - \cos^2 A = 1 - 0{\cdot}3078 = 0{\cdot}6922$$

$$\sin A = \sqrt{0{\cdot}6922} = \underline{0{\cdot}8320}$$

Also, $\quad \operatorname{cosec} A = \frac{1}{\sin A} = \frac{1}{0{\cdot}8320} = \underline{1{\cdot}2019}$

$$\cot A = \frac{1}{\tan A} = \frac{1}{1{\cdot}5} = \underline{0{\cdot}6667}$$

After all the calculation, we may verify that

$$\operatorname{cosec}^2 A - \cot^2 A = 1{\cdot}4447 - 0{\cdot}4444 = 1{\cdot}0003$$

a cumulative error of 3 in 10,000 only.

A comparison of the two methods should convince the student that the triangle one is less trouble for numerical work, but he should *not* conclude from that that the formulae are not useful. Their sphere of usefulness is a different one. Of course, for numerical instances, there is the third alternative of using tables, but unless care is taken of the minutes—and fractions of minutes—there are likely to be cumulative small errors whose magnitude is difficult to determine, due to successive approximations at each reference to a table.

EXERCISE No. 27

1. Find all the other ratios in the following instance: (*a*) $\cos A = 0{\cdot}9$, (*b*) $\tan A = 3$, (*c*) $\sec A = 1{\cdot}05$.
2. Denoting $\sin A$ by s, find formulae for the other ratios in terms of s.
3. If $\sin A = 0{\cdot}3$, find the values of $2 \sin A \cos A$, and of $\cos^2 A - \sin^2 A$. By means of tables, verify that these are the sine and cosine of double the angle A, respectively.

Answers to Exercise No. 25

1. 13·4 yd. = 40·2 ft.
2. 70° 38′, 28° 11′.
3. 23·2 yd., 153·3 yd.
4. 10·71 in., 80·33 sq. in.
5. 24·4 ft.
6. 94·09 ft., 85·86 ft.
7. 71·8 yd. = 215·4 ft.
8. 40·9 yd. = 122·7 ft.
9. 1,160 ft., 204 ft.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XVI

Machine Cut Gearing. Proceeding, from the last lesson, with the study of spur gear teeth we may now give our attention to the case of the practical as distinct from the academical drawing. Taking, as an example, the construction of two wheels required to reduce the speed of rotation of the shafts to which they are fixed from 120 to 80 r.p.m., we may assume full freedom with regard to the available space and endeavour to produce the smallest wheels within the limits of permissible stress.

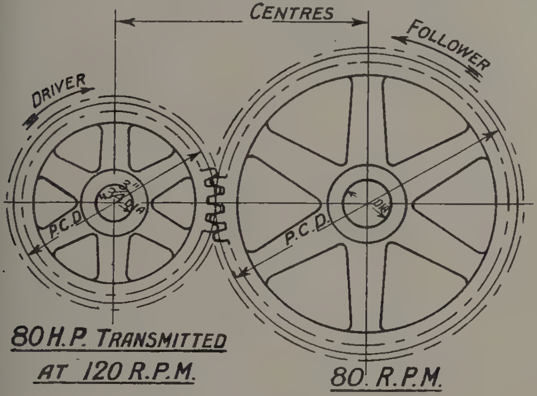


Fig. 1

Referring to Fig. 1, the driver is secured to a shaft of $3\frac{3}{4}$ in. diameter transmitting 80 h.p., and we may proceed to find the *p.c. diameter* of the wheels, together with the diameter of the shaft supporting the follower, and so complete the dimensioning of this basis plan.

Limiting Values. The above example, in design, presents the common feature of possessing a number of variable dimensions relating the strength of the detail to the stresses induced under load. In such cases, it is convenient to consider the limits to which any of the factors may be subjected, and we may readily ascertain the limiting values in the above problem by bringing the various factors under review.

Let h.p. = horse-power transmitted by the driver.
,, N = speed of the driver in revolutions per minute,
Let T = twisting moments transmitted through driver in lb.-in.
,, p = circular pitch of teeth in inches.
,, w = width of teeth in inches.
,, h = total height of tooth above the root in inches.
,, D = pitch circle diameter of teeth in inches.
,, n = number of teeth in wheel.
,, W = tooth load at pitch point in a line tangential to the pitch circle in lb.
,, f = maximum stress induced in the teeth in lb. per sq. in.
,, d = diameter of wheel shaft in inches.
,, K = constant depending upon the value of n.

Then, considering the condition of one tooth taking all the load at the free end, we have

$$T = \frac{33{,}000 \text{ h.p.} \times 12}{2.\pi.N.} = \frac{63{,}000 \text{ h.p.}}{N}$$

$$W = \frac{T}{\frac{D}{2}} = \frac{2T}{D} \quad . \quad . \quad . \quad (1)$$

Reference to Fig. 2 shows that the approximate maximum bending moment on the tooth

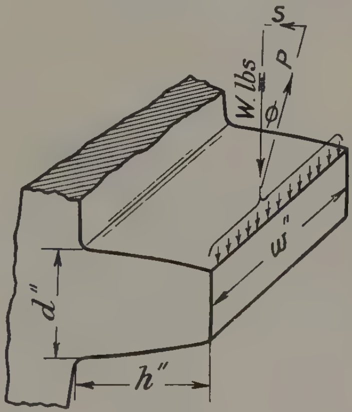


Fig. 2

$= B.M. = W \times h$ lb.-in., and the moment of resistance offered by the tooth $= \frac{w.d^2}{6} f$

$$\text{then } \frac{w.d^2.f}{6} = W.h,$$

or, substituting ·5 . $K . p$. for d and ·6836p for h,

$$\frac{w.(\cdot 5\,Kp)^2 f}{6} = = \cdot 6836 . W . p.$$

$$w . K^2 . p . f = 16\cdot 41 W.$$

substituting from (1),

$$w . K^2 p . f . = \frac{2 . T.}{D} \times 16\cdot 41$$

$$\text{but } D = \frac{np}{\pi}$$

$$\therefore\ n . w . K^2 . p^2 . f . = 103\cdot 0 T. \qquad . \qquad (2)$$

Of the above factors we find: The value of T is fixed by the specified power and speed of the machinery. The value of f chiefly depends upon the material used, but it is also dependent upon the tooth speed at the pitch point. The constant K is introduced to allow for the root thickness varying from the value ·5p. Referring to Fig. 3, the tooth forms shown are for teeth having the same circular pitch but attached to wheels of varying pitch circle diameter. Table I gives the ratio $\frac{T}{t}$ for wheels having various numbers of teeth and a pressure angle of $14\frac{1}{2}°$.

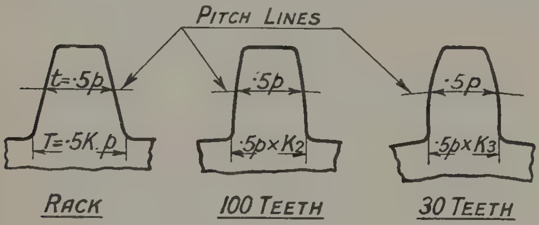


FIG. 3

The limiting values in the above example are for the factors n, f, and w, and if we are endeavouring to obtain the smallest wheel commensurate with a reasonable strength we ought, in fixing the wheel dimensions, to adopt values up to the limit.

Taking 32 as a minimum value for n, 3p as a maximum value for w, and tentatively taking a value of 4,000 for f (making the wheel of cast iron), we may proceed to find the minimum value for p. Substituting in (2) above, we have

TABLE I
(REFER TO FIG. 3)

No. of Teeth	30	40	60	80	100	150	200	Rack
$K = \frac{T}{t}$ Press Angle = $14\frac{1}{2}°$	1·07	1·11	1·17	1·23	1·25	1·30	1·32	1·37

$$32 \times 3p . K^2 . p^2 . f = 103\cdot 0 T$$

$$p^3 = \frac{1\cdot 078 . T}{K^2 f}$$

$$\text{or approximately, } p = \left(\frac{T}{K^2 f}\right)^{\frac{1}{3}}$$

$$\text{For the above example } T = \frac{63{,}000 \times 80}{120}$$

$$= 42{,}000,$$

$$\text{and } p = \left(\frac{42{,}000}{1\cdot 078^2 \times 4000}\right)^{\frac{1}{3}} = 2\cdot 083 \text{ in.}$$

With the above value of p, the speed of the teeth at the pitch point will be

$$= \frac{n . p . N}{12} = \frac{32 \times 2\cdot 08 \times 120}{12} = 666 \text{ ft./min.}$$

Trial and Error. Table II gives values of working stresses, for various materials, at varying tooth speeds. Interpolating the values in the above table we find that the working stress for cast-iron teeth at a speed of 666 = 3,800 lb.-sq. in. This value is sufficiently near to the above tentative value of 4,000 lb.-sq. in. to allow us to retain 2·08 in. as the minimum circular pitch.

In the event of the allowable stress being well below the tentative value of f used in equating for p, it would be necessary to substitute a smaller value for f in the equation and

TABLE II
MAXIMUM VALUES OF f AT VARIOUS TOOTH SPEEDS

Tooth speed in feet per minute	100 or less	200	400	600	800	1,000	1,500	2,000	2,500
Cast iron . . .	8,000	6,000	4,800	4,000	3,400	3,000	2,300	1,800	1,500
Cast steel . . .	20,000	15,000	12,000	10,000	8,500	7,500	5,750	4,500	3,750
Forged steel . .	24,000	18,000	14,000	12,000	10,000	9,000	7,000	5,000	4,500
Bronze . . .	11,000	8,500	6,500	5,500	5,000	4,000	3,000	2,500	2,000

so find a new circular pitch. Although the above method is of the *trial and error* order, it will be found that, with experience, a second calculation is seldom necessary.

For convenience in the workshop, the circular pitch p is converted into a value called the diametral pitch.

The diametral pitch $= \dfrac{\pi}{\text{circular pitch}}$ and gives the number of teeth per inch of the pitch circle diameter. Further, for economy in the production of gear cutting tools standard values of diametral pitch are adopted, viz., 1, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{3}{4}$, 2, ... The diametral pitch equivalent to the minimum circular pitch as found for the above wheel is an odd value, and, taking the nearest standard diametral pitch of $1\frac{1}{2}$, we find the corresponding circular pitch = 2·09 in.

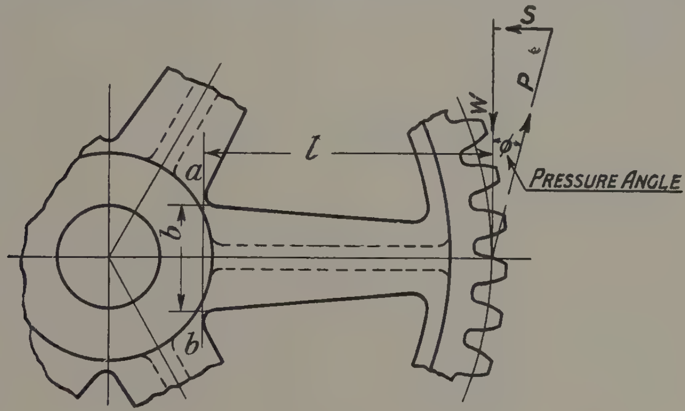
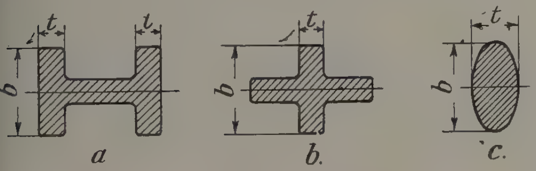


FIG. 4



a b. c.

FIG. 5

From this value we find

$$D = \frac{32 \times 2{\cdot}09}{\pi} = 21{\cdot}28 \text{ in.}$$

$w = 3 \times 2{\cdot}09 = 6{\cdot}27$ in., say, $6\frac{1}{4}$ in.
Tooth addendum $= {\cdot}3183 \times 2{\cdot}09 = {\cdot}6651$ in.
,, ,, $= {\cdot}3683 \times 2{\cdot}09 = {\cdot}7696$ in.

Proceeding to the wheel rim we may adopt a proportion of thickness of rim equal to the tooth thickness at the root. This proportion would give an approximate value of $\cdot 5 \times 2{\cdot}09 = 1{\cdot}045$, say, $1\frac{1}{8}$ in. The hub diameter may be found from the greater of the following empirical rules—

Diameter of hub $= 1{\cdot}6d$ or $d + 1{\cdot}6p$.
i.e. ,, ,, $= 1{\cdot}6 \times 3{\cdot}75 = 6$ in.
or ,, ,, $= 3{\cdot}75 + 1{\cdot}6 \times 2{\cdot}09$
$= 7{\cdot}08$ in., say, 7 in.

We have now to join the wheel rim to the hub. Experience would suggest the most suitable number of arms for the above wheel diameter to be six, and, distributing the tooth load equally between the arms, we find the load to be taken by each arm $= \dfrac{W}{6} = \dfrac{2T}{6D}$

$$= \frac{42{,}000}{3 \times 21{\cdot}28} = 658 \text{ lb.}$$

Referring to Fig. 4, it is evident that the maximum depth of each arm at the hub will be equal to the distance between the points of intersection of the arm centre lines and the hub circle, and that the section *ab* will be one of greatest stress. Taking moments about this section, we have the bending moment

$$\text{B.M.} = \frac{W \times l}{\text{No. of arms}} = 658 \times 7{\cdot}65 = 5034 \text{ lb.-in.}$$

We have now to choose a suitable section for the arm at *ab*, having a moment of resistance equal to the above bending moment when subjected to a reasonable stress. Fig. 5 shows three common types of arm section, and we need not consider the respective merits of the types until we have found the dimensions required for each type to provide the required moment of resistance. We have the approximate moment of resistance

$$\text{of type } a = \frac{2t\,.\,b^2.}{6} f \text{ lb.-in.}$$

$$,, \quad b = \frac{tb^2}{6} f \quad ,,$$

$$,, \quad c = \frac{tb^2}{10{\cdot}2} f \quad ,,$$

Allowing a working value for $f = 1600$ lb.-sq. in.

$$\text{for type } a, \ t = \frac{3.\text{B.M.}}{b^2.f.} = \frac{3 \times 5034}{3{\cdot}5^2 \times 1600} = {\cdot}77 \text{ in}$$

$$,, \quad b, \ t = \frac{6 \times 5034}{3{\cdot}5^2 \times 1600} = 1{\cdot}54 \text{ in.}$$

$$,, \quad c, \ t = \frac{10{\cdot}2 \times 5034}{3{\cdot}5^2 \times 1600} = 2{\cdot}62 \text{ in.}$$

The values of t for types b and c are disproportionate, and, consequently, we may decide to fit arms similar to type a, Fig. 5.

We may now complete the drawing (Fig. 7) to the dimensions as determined above and, observing the practical nature of the drawing, show only a few teeth in the region of the arc of contact. Further, in drawing the tooth outline, we may adopt the approximate method suggested in Fig. 6, where the tooth profile is produced by an arc of constant radius.

Much of the above work may be utilized in the design of the wheel gearing with the above driver. Using the same notation for the two wheels with the addition of the suffix 1, for the following wheel.

Then $p_1 = p.$
$w_1 = w.$

and neglecting the inefficiency of the gear

$$T_1 = \frac{N}{N_1} \times T = \frac{120}{80} \times 42{,}000 = 63{,}000 \text{ lb.-in.}$$

Also allowing the same working stress in the shafts,

$$d_1 = \left(\frac{T_1}{T}\right)^{\frac{1}{3}} d = \left(\frac{63{,}000}{42{,}000}\right)^{\frac{1}{3}} \times 3{\cdot}75$$

$$d_1 = 1{\cdot}145 \times 3{\cdot}75 = 4{\cdot}29 \text{ in., say, } 4\tfrac{1}{4} \text{ in.}$$

$$D_1 = \frac{N}{N_1}\,.\,D = \frac{120}{80} \times 21{\cdot}28 = 31{\cdot}92 \text{ in.}$$

$$n_1 = \frac{N}{N_1} n = \frac{120}{80} \times 32 = 48.$$

It will be observed that, in obtaining equation (2), the compressive stress due to the smaller component force has been neglected. The increase in the above nominal stress, resulting from a consideration of the smaller component force, is more than compensated by the fillet at the root of the tooth.

A tooth form of thicker proportion is finding increasing favour among engineers, and has, in addition to smaller addendum and dedendum, a pressure angle of 20°.[1]

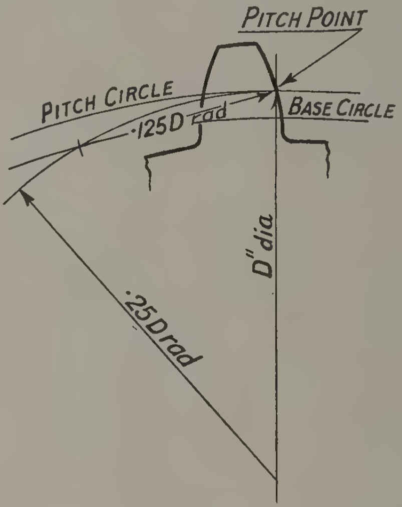


FIG. 6

Table III gives the corresponding values of K for this form of tooth having an addendum of $\cdot 25p$ and dedendum of $\cdot 3p$. Given the same tooth load and pitch the bending moment will be correspondingly smaller for the above tooth form than for the Brown and Sharpe standard. The approximate B.M. $= \cdot 55\, p.W.$ lb.-in., and

$$p = \left(\frac{27{\cdot}65\,.\,T.}{n.\,K^2.\,f.)}\right)^{\frac{1}{3}}$$

TABLE III

Value of n .	30	40	60	80	100	150	200	Rack
,, ,, K .	1·21	1·26	1·33	1·37	1·39	1·42	1·43	1·44

EXERCISE

Proceed with the above calculations for the *following* wheel, and make a working drawing of same similar in style, to Fig. 7.

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[1] See article in Engineering by W. E. Wright, B.Sc. on 10th September, 1926.

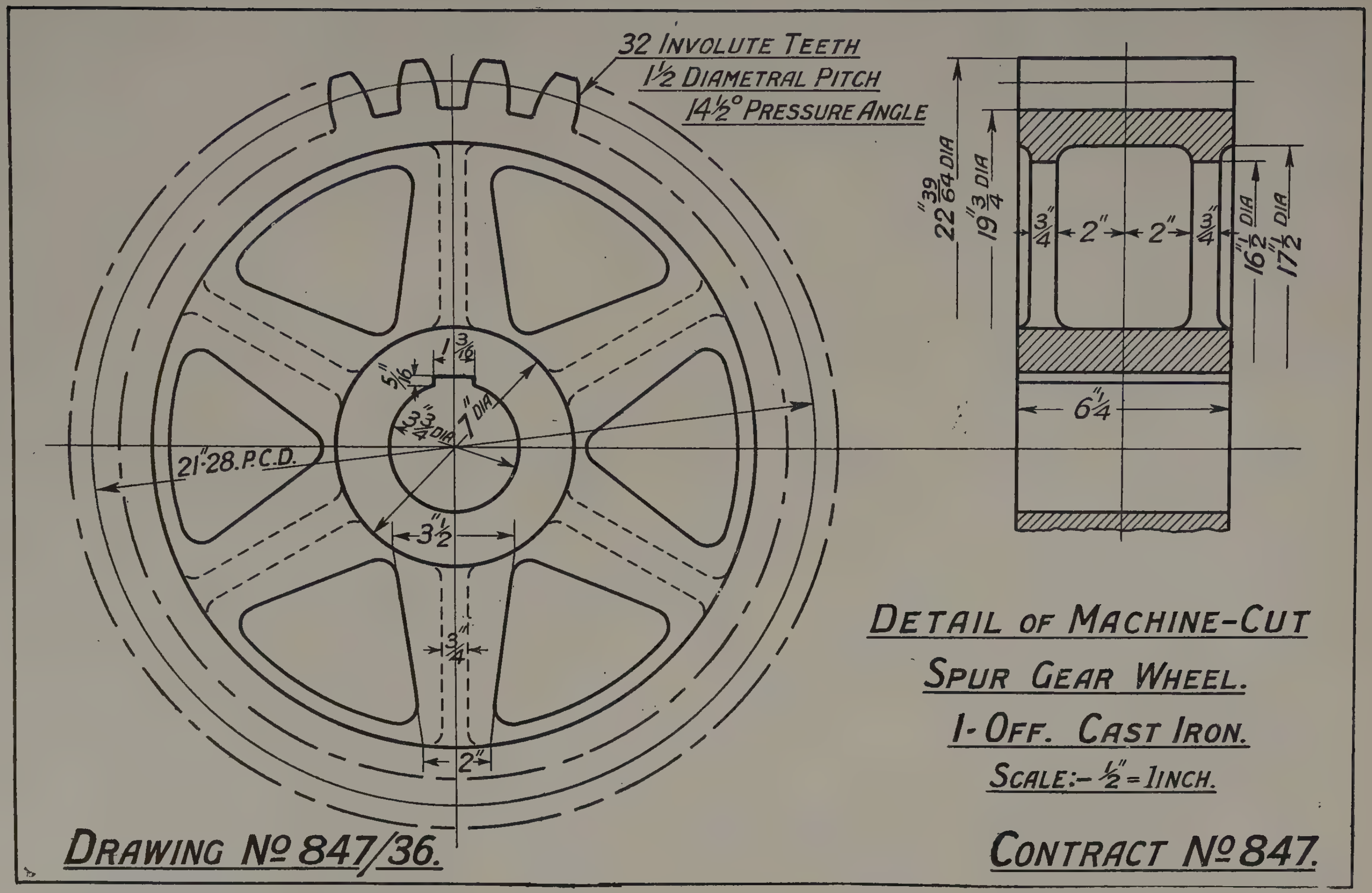

32 INVOLUTE TEETH
1½ DIAMETRAL PITCH
14½° PRESSURE ANGLE
22 39/64" DIA
19 3/4" DIA
3/4"
2"
2"
3/4"
16 1/2" DIA
17 1/2" DIA
6 1/4"
1 3/16
5/16"
3 3/4" DIA
7" DIA
21"28. P.C.D.
3 1/2"
3/4"
2"
DETAIL OF MACHINE-CUT
SPUR GEAR WHEEL.
1-OFF. CAST IRON.
SCALE:- 1/2" = 1 INCH.
DRAWING Nº 847/36.
CONTRACT Nº 847.

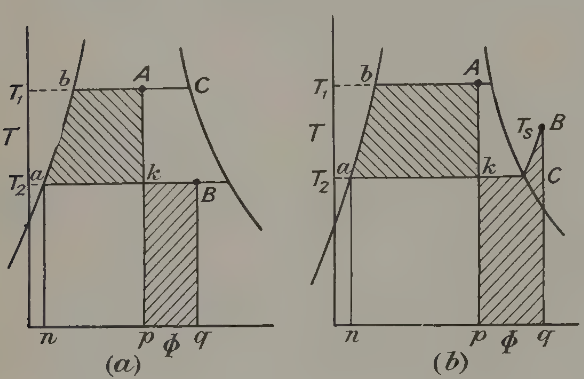
FIG. 7

THE THEORY OF HEAT ENGINES

By E. B. Cole, B.Sc.

LESSON X

Gain of Entropy Due to Throttling. We have already seen that when steam is throttled the total heat after is the same as before throttling. In Fig. 24 (*a*), if *A* represents the initial dryness and temperature of the steam on the T–ϕ chart, then the total heat, reckoned above that of water at T_2, possessed by the 1 lb. of steam is given by area *nabAp*. Then if, after throttling, the point *B* represents the condition, the area *naBq* represents the total heat and must, therefore, be equal to *nabAp*.



(*a*) (*b*)

FIG. 24

In other words, the two shaded areas are equal, and what has occurred is that the heat energy as represented by area *abAk* has been converted into kinetic energy of the steam jet which, in turn, has been destroyed and reappears as heat, represented by the area *pkBq*. As shown, the steam is still wet after throttling.

Fig. 24 (*b*) shows the same effect, but the heat liberated was sufficient to superheat after throttling. Thus, the gain in entropy is *pq* in each case, and it will be left as an exercise for the student to show that this gain is

$$(a)\quad pq = \frac{T_1 - T_2}{T_2}\left(1 + \frac{x_1 L_1}{T_1}\right) - \log_e \frac{T_1}{T_2}$$

$$(b)\quad pq = \frac{L_2}{T_2} + k_p \log_e \frac{T_s}{T_2} - \left[\log_e \frac{T_1}{T_2} + x_1 \frac{L_1}{T_1}\right]$$

In a steam turbine, steam is similarly expanded through a nozzle, and a portion of its heat energy as represented by the area *abAk* is in a similar manner converted into kinetic energy, the steam issuing as before as a high-velocity jet. But this jet is not uncontrolled and destroyed in eddies; instead it is allowed to do external work by being directed on to the blades of the turbine rotor.

Thus, the velocity energy is not reconverted into heat by molecular friction or eddies, and therefore there is no increase in entropy after the expansion. Hence, as external work is also done, this expansion may be termed *adiabatic*. As defined, an adiabatic change is isentropic, i.e takes place at constant entropy. We are now in a position to discuss an ideal heat cycle using water and steam as the working fluid, namely—

The Rankine-Clausius Cycle, so named, since it was devised by both Rankine and Clausius independently. This cycle is of the utmost importance since the steam-engine works upon a cycle approaching it, while the steam turbine works almost exactly upon it.

Let us examine the cycle in regard to 1 lb of water enclosed in a cylinder fitted with a movable piston. Fig. 25 gives the *PV* diagram.

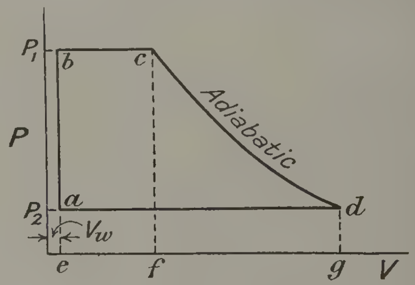


FIG. 25

1. At "*a*" we have 1 lb. of water at pressure P_2 and temperature T_2, which is the lower limit. From $a \rightarrow b$ the pressure of the water is raised to a higher level P_1, and heat is supplied to raise the temperature to T_1, being that corresponding to P_1 at which evaporation will start. This operation is *not* reversible, since heat is being supplied while a change of temperature occurs.

In practice, the heat is not supplied to the water inside the engine cylinder, but in a separate container, namely, the boiler. The feed pump raises the pressure of the water, while the heat is supplied in the boiler and, perhaps, partly by feed heaters, as the water is pumped into the boiler.

2. The water is evaporated at the temperature T_2, corresponding to P_2, the piston moving from b to c. The evaporation actually takes place in the boiler, but, since this is in communication with the cylinder, when the pound of steam is admitted, the piston would move to c. If the dryness of the steam formed is x_1, then the heat supplied is x_1L_1. This operation is reversible.

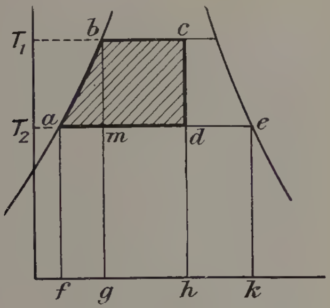


FIG. 26

The volume represented by c will be x_1V_{s1}, where V_{s1} is the specific volume at P_1.

3. The steam is then expanded adiabatically to the lower pressure P_2 doing work on the piston, the volume increasing to d, being equal to x_2V_{s2}—where x_2 is the dryness fraction after the expansion, and V_{s2} the specific volume at P_2. This operation is reversible.

4. Condensation of the steam at constant temperature T_2 now takes place, the piston returning to its original position at "a."

In practice this operation, i.e. the condensation of the steam, also occurs in a separate chamber, i.e. the condenser, connected to the cylinder.

The heat rejected will be x_2L_2, and the operation is reversible.

This completes the cycle.

The shaded area $abcd$ represents the work done per pound, and this may be evaluated as follows—

Area $abcd$ = area $ebcf + fcdg - eadg$.

Now neglecting the volume V_w of 1 lb. of water, which is very small compared with that occupied as steam,

$$ebcf = P_1{\cdot}x_1V_{s1}$$
$$eadg = P_2{\cdot}x_2V_{s2},$$

and assuming the adiabatic curve to be of the type $PV^n = C$—we saw in the last lesson how to determine the value of n—and hence

$$fcdg = \frac{P_1.x_1V_{s1} - P_2.x_2V_{s2}}{n - 1}$$

Hence on substitution,

W = work done = area $abcd$

$$= \frac{n}{n - 1}[P_1x_1V_{s1} - P_2x_2V_{s2}] \text{ ft.-lb. per lb.}$$

The heat supplied per lb.

$$= (h_1 - h_2) + x_1L_1 \text{ heat units}$$
$$= (T_1 - T_2) + x_1L_1$$

∴ thermal efficiency

$$= \frac{W}{J[(T_1 - T_2) + x_1L_1]}$$

This evaluation from the PV diagram is distinctly laborious, and the result may be obtained directly as follows—

We have—

Work done per cycle = heat supplied − heat rejected
Heat supplied as above in stages (1) and (2) = $(T_1 - T_2) + x_1L_1$ heat units per lb.
Heat rejected, stage (4) = x_2L_2 units per lb.
∴ work done/lb. = $T_1 - T_2 + x_1L_1 - x_2L_2$
∴ thermal efficiency = $\dfrac{T_1 - T_2 + x_1L_1 - x_2L_2}{T_1 - T_2 + x_1L_1}$

The value of x_2 is at once determinable from the $T - \phi$ chart, and the cycle is much more clearly investigated by means of this chart than with the PV diagram, since from the latter no indication of the quality of the steam is obtained. In Fig. 26, the Rankine cycle is represented on the $T - \phi$ chart, with lettering to correspond to that of the PV diagram—

1. $a \rightarrow b$, we have 1 lb. of water at "a" at a temperature T_2 corresponding to the pressure P_2. Sensible heat is supplied to raise temperature to T_1, and at the same time the pressure is raised to P_1, such that evaporation may now take place at temperature T_1.

Area $fabg$ = sensible heat supplied
$= T_1 - T_2$

change of entropy $= am = \log_e \dfrac{T_1}{T_2}$

2. $b \rightarrow c$, evaporation at constant temperature T_1.

Area $gbch = x_1L_1 =$ latent heat supplied

Change of entropy $= bc = \dfrac{x_1L_1}{T_1}$

3. $c \rightarrow d$, adiabatic expansion to lower temperature T_2. No heat is supplied or rejected, therefore there is no change of entropy. Dryness fraction at end of expansion we have seen to be

$$x_2 = \frac{ad}{ae} = \frac{\log_e \dfrac{T_1}{T_2} + x_1 \dfrac{L_1}{T_1}}{L_2/T_2}$$

4. $d \rightarrow a$, condensation at constant temperature T_2.

Area $fadh =$ heat rejected $= x_2L_2$.

Thus—

Work done/lb. per cycle = heat supplied − heat rejected

$= \text{area } fabch - \text{area } fadh$

$= \text{area shaded } abcd$

$= (T_1 - T_2) + x_1L_1 - x_2L_2$

or substituting for x_2

$$= (T_1 - T_2) + x_1L_1 - T_2 \log_e \frac{T_1}{T_2} - T_2x_1 \frac{L_1}{T_1}$$

$$= (T_1 - T_2) + x_1 \left[\frac{T_1L_1}{T_1} - \frac{T_2L_1}{T_1}\right] - T_2 \log_e \frac{T_1}{T_2}$$

$$= (T_1 - T_2)\left[1 + x_1 \frac{L_1}{T_1}\right] - T_2 \log_e \frac{T_1}{T_2}$$

$\therefore$ efficiency

$$= \frac{(T_1 - T_2)\left[1 + x_1 \dfrac{L_1}{T_1}\right] - T_2 \log_e \dfrac{T_1}{T_2}}{(T_1 - T_2) + x_1L_1}$$

This efficiency will be less than the ideal $\dfrac{T_1 - T_2}{T_1}$, since the step ab of the cycle is not reversible, heat being taken in during a change in temperature.

Rankine Cycle for Superheat Steam. Fig. 27 shows the cycle in which the water is completely evaporated, and the steam superheated to a temperature T_s above the temperature T_1, corresponding to the pressure P_1.

The *heat supplied* = area *fabnch*

$= fabg + gbnp + pnch$

$= T_1 - T_2 + L_1 + k_p(T_s - T_1)$ heat units/lb.

The *heat rejected* = area *fadh* $= x_2L_2$

$$= T_2\left(\log_e \frac{T_1}{T_2} + \frac{L_1}{T_1} + k_p \log_e \frac{T_s}{T_1}\right)$$

Thus, the work done per cycle per lb. = area *abcd*

$$= (T_1 - T_2)\left(1 + \frac{L_1}{T_1}\right) + K_p(T_s - T_1) - T_2\left(\log_e \frac{T_1}{T_2} + k_p \log_e \frac{T_s}{T_1}\right)$$

It will be realized that, if H = total heat of 1 lb. of steam before adiabatic expansion, and H_2 that after expansion, the work done per pound of steam on the Rankine cycle, as deduced above, is $H_1 - H_2$. This difference in the total heats which can thus be converted into mechanical work is termed the *Rankine Heat Drop*.

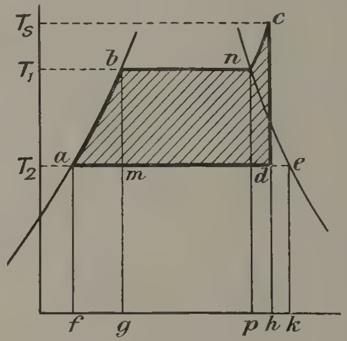


FIG. 27

Let us work an example.

An engine is supplied with superheated steam at 120 lb. per sq. in. absolute ($t = 341°$ F.; $L = 873{\cdot}7$ B.Th.U.), and exhausts at 4 lb.-sq. in. absolute ($t = 153°$ F., $L = 1007$). Taking the mean specific heat of the steam as 0·5, find the superheat that must be given in order that after adiabatic expansion over this pressure range the steam may be just dry and saturated.

An engine works on the Rankine cycle with this degree of superheat, and between the same pressures. Find the work done per pound, the thermal efficiency and the theoretical steam consumption in pounds per horse-power hour.

(*Continued on page* 799)

THE FOSTER SUPERHEATER

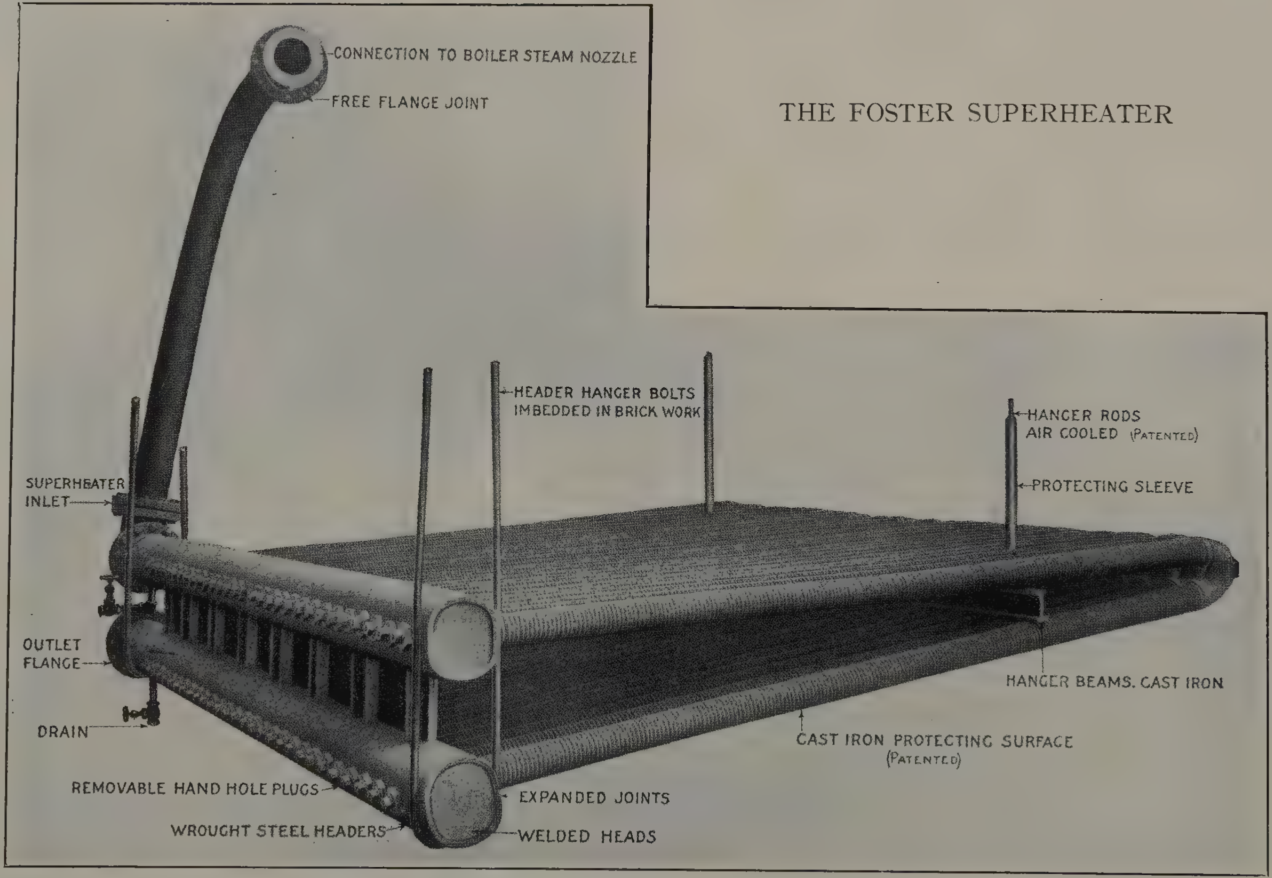

CONNECTION TO BOILER STEAM NOZZLE
FREE FLANGE JOINT
HEADER HANGER BOLTS IMBEDDED IN BRICK WORK
HANGER RODS AIR COOLED (PATENTED)
PROTECTING SLEEVE
SUPERHEATER INLET
OUTLET FLANGE
HANGER BEAMS. CAST IRON
DRAIN
CAST IRON PROTECTING SURFACE (PATENTED)
REMOVABLE HAND HOLE PLUGS
EXPANDED JOINTS
WROUGHT STEEL HEADERS
WELDED HEADS

Fig. 38. Showing Method of Suspending the Superheater in a Vertically Baffled Horizontal Water-tube Boiler

STEAM BOILERS AND THEIR ACCESSORIES

By Percy E. Rycroft, M.B.E., M.I.E.E.

LESSON VI

SUPERHEATERS

The function of a superheater is to increase the temperature of the steam above that due to its pressure, in order to compensate for temperature drop, due to radiation and the heat lost during its performance of work by expansion in an engine or turbine.

As in the case of economizers, where heat is recovered from the boiler flue gases to pre-heat the boiler feed, the superheater is also placed in the path of the furnace gases, but usually in a position nearer to the fire zone.

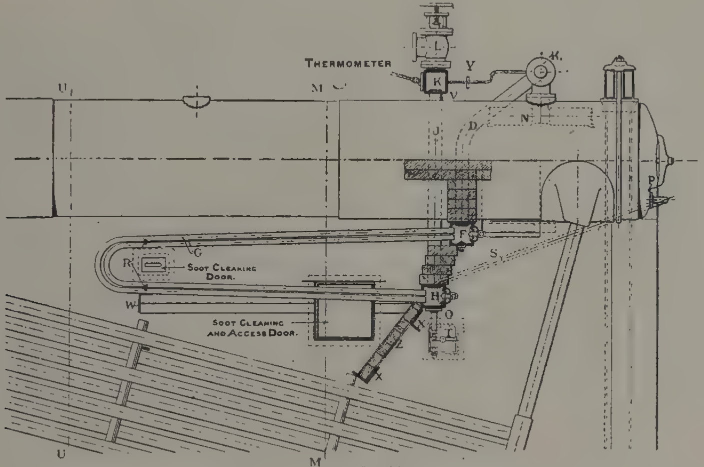


FIG. 32. Babcock & Wilcox Integral Superheater

(Single Loop)

The actual position of the superheater will naturally vary with the intensity of superheat desired, and in the majority of water-tube boilers it is placed between the first and second "pass," where a gas temperature of about 1,100 or 1,200° F. may be expected and a final steam temperature of about 600° F. may be anticipated.

To provide still higher temperatures to meet the high turbine efficiencies now demanded involving final steam temperatures of 700 to 750° F., the "inter-deck" superheater is being increasingly used. This arrangement divides the boiler tube nest into two sections, and leaves a space for the superheater between them so that it is subjected to a still hotter fire zone.

At one time separately fired superheaters had some vogue. By this arrangement all the steam produced by a battery of boilers was passed through one or more large superheaters, each being fitted with its own coal-fired furnace. It had the advantage of providing a very ready means for varying the final temperature within very wide limits, but it was almost equally easy for the temperature to rise too high, especially under conditions of light load, and burnt tubes often resulted.

To-day practically all superheaters form an integral portion of the complete steam-raising unit, the superheater being protected by at least a few rows of boiler water tubes.

The question of varying the amount of superheat imparted to the steam is sometimes raised, particularly in the older installations where a mixed plant, some of which is incapable of withstanding the higher temperature, has occasionally to be run.

If the variation required is only slight, say a maximum of 50° F., it is possible to obtain the desired result by means of a mixing valve, i.e. a certain quantity of saturated steam is admitted direct from the boiler drum to the superheater valve connection and there mixes with the superheated steam.

In some cases, baffle dampers have been introduced into the furnace gas passages, in order to short circuit a portion of the gases and so vary the temperature of the superheater chamber, but there are difficulties in providing frames and dampers capable of withstanding the temperature to which they would be subjected under these conditions.

Perhaps the best method of steam temperature regulation so far designed is by means of an "attemporator," whereby the steam or such proportion as may be necessary for the purpose is passed through a gilled pipe or coil placed inside the boiler drum below water level. The temperature of the boiler water being necessarily less than that of the superheated steam, it follows that heat will be given up by the latter and pass into the boiler feed, so that while the desired temperature regulation will be obtained practically no heat loss will be incurred over the whole plant.

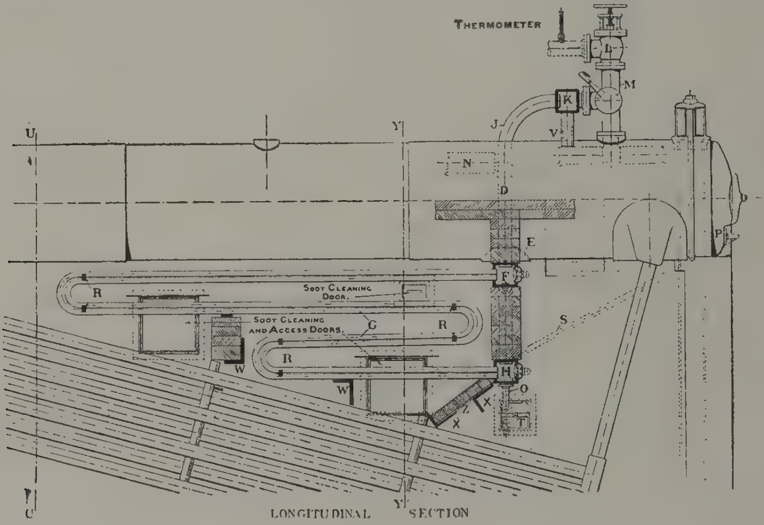


FIG. 33. BABCOCK & WILCOX INTEGRAL SUPERHEATER

(Double Loop)

The advantages of superheated steam are particularly evident in connection with steam turbines, where for the first 100° of superheat a saving of 1 per cent in fuel may be anticipated for each 10° of temperature rise. Beyond the first 100° the ratio of saving is slightly less, but at 250° F. superheat, it approaches 20 per cent.

A further advantage of dry steam is that wear on the turbine blading, due to the presence of moisture, is eliminated.

For reciprocating plant with cast-iron cylinders and extensive sliding surfaces it is not possible to use such high temperatures as with turbines—where 700 to 750° F. is now common practice—but moderate temperatures of, say, 450 to 500° F. are quite permissible and are of the greatest advantage. Apart from a lesser quantity of steam being required, due to its volume not being reduced by rapidly returning to water by condensation on the cylinder walls, there is the further advantage of more efficient lubrication, as dry steam has no tendency to wash away the cylinder oil.

In essentials all superheaters are designed on similar principles, and consist of a number of small diameter steel tubes proportioned, as to length, to expose a considerable total heating surface—depending on the degree of superheat required—to the furnace gases, and the examples illustrated in this section are typical of modern practice.

The Babcock and Wilcox Superheater. This apparatus is supplied as an integral portion of the boiler, and is placed either immediately above the boiler tubes or—where very high temperatures are required—between two horizontal sections.

No part of the superheater is subjected to direct impact with the fire, and access to the apparatus for internal and external cleaning is provided through suitable inspection doors.

The superheater consists of solid drawn steel tubes, 1¼ in. diameter, bent to form a single or double loop, in accordance with the total amount of heating surface necessary, and expanded into steel "headers" or manifolds. Access is obtained to the ends of the tubes through suitable hand holes provided in the manifolds, and closed by steel caps.

Expansion is provided for by the tubes being quite free at the loop end, and by the fact that the upper and lower manifolds are not rigidly connected to each other.

The saturated steam passes from the boiler drum to one manifold, through the superheater tubes into the other from which it is delivered to the valve outlet.

As a precaution against the possibility of the tubes becoming overheated during the initial stages of raising steam in the plant, i.e. before steam has started flowing through the superheater, a special arrangement of combined looding valve and drain valve is supplied.

By this apparatus the superheater can be lled with water until such time as the boiler begins to generate steam, when it can be conveniently drained away and the superheater llowed to function.

For the lower superheats up to, say, 100 to 150° F. above the saturated steam temperature, the single loop design is usually employed, while beyond this the double loop type is installed.

Although, as has already been stated, it is now general practice for the superheater to form an integral portion of the complete steam-raising unit, conditions may arise in which an independently fired superheater is called for, and a typical arrangement adopted by this firm is illustrated in Fig. 35.

A particularly recent evolution of the superheater is now being supplied by Messrs. Babcock to a number of the super electric power stations.

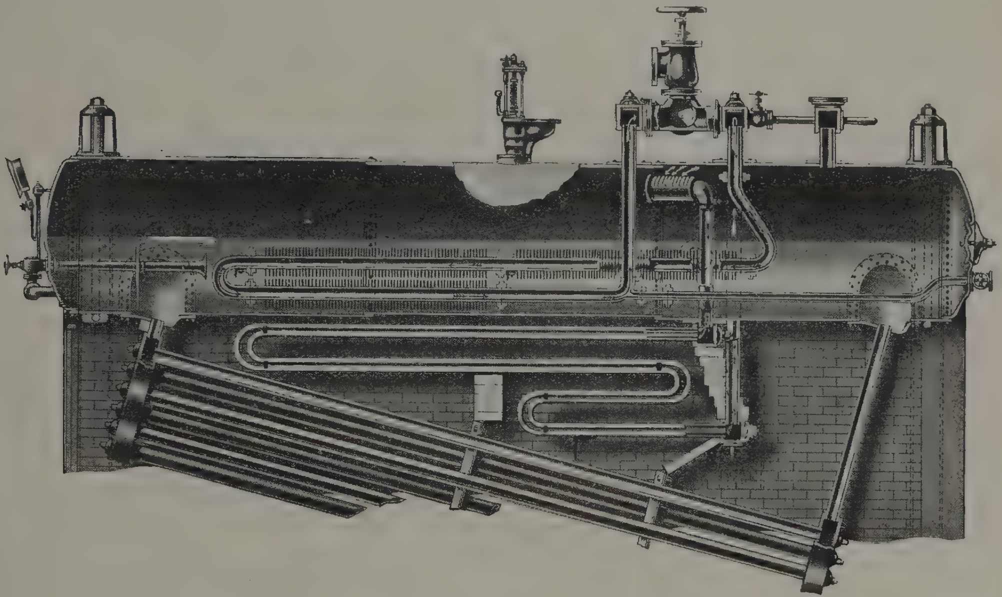


Fig. 34. Babcock & Wilcox Attemporator

It takes the form of a reheater for low-pressure steam which is returned to it from the high pressure turbine discharge before it passes to the lower pressure turbine stages to complete its expansion. In a plant now working, the initial total temperature of the steam is 650° F. at 450 lb. pressure. After partial expansion the steam returns to the reheater at about 65 lb. pressure, and a total temperature of about 350° F. This steam is reheated back to about 500° F., and returned to the low pressure end of the turbine to complete its work.

The reheater is embodied in a standard boiler, and the elements are protected against direct impact with the fire by several rows of boiler water tubes.

Foster Superheaters. The Foster superheater is installed either as an integral portion of the boiler or as a separate unit.

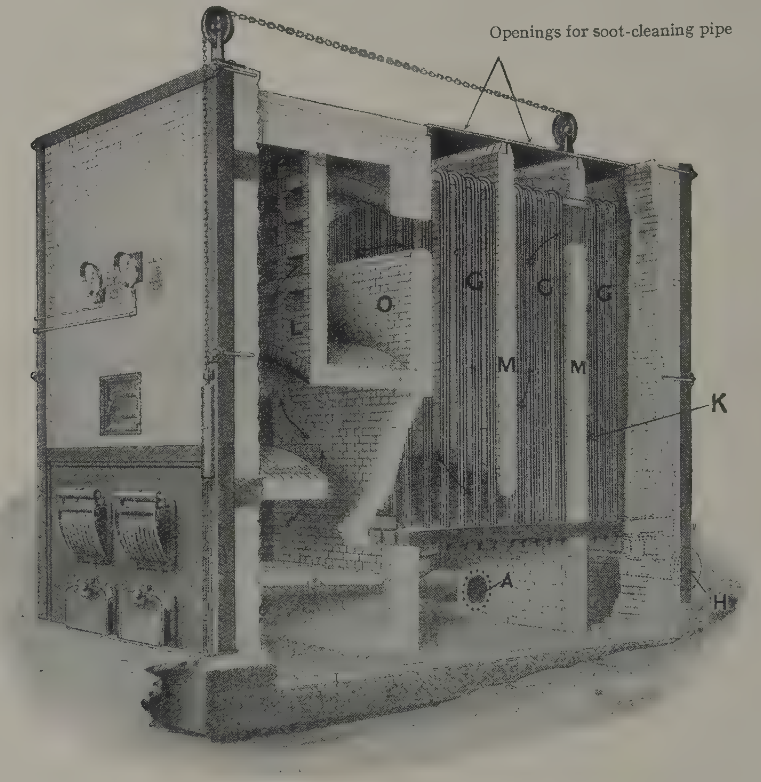


Fig. 35. Babcock & Wilcox Independently-fired Superheater

A. Main steam connecting pipe.
G. Superheater tubes.
H. Steam delivery pipe.
K. Flue damper.
L. Perforated firebrick wall.
M. Flame baffles.
O. Muffling chamber.

In its standard form it consists of a series of small diameter straight steel tubes expanded into steel manifolds or collecting boxes, and enclosed over their entire length by cast-iron rings or gills.

These rings not only protect the steel tubes from external corrosion, but add appreciably to their strength.

The gills are primarily adopted to provide increased heat absorbing surface which has, in addition, a stabilizing action or fly-wheel effect in equalizing the superheat temperatures under varying conditions of load.

The straight lengths of steel tube are connected at their return ends by semicircular steel connectors, and the whole is supported by girders carried by air-cooled suspension rods.

A special feature of this superheater is the provision of an inner core consisting of a small diameter tube sealed at each end, and centrally spaced in the superheater tube by suitable spacing pieces. The object of this arrangement is to compel the steam to flow in a thin annular stream quite close to the tube surfac in order to ensure a maximum heat transmission.

Access to the interior of the tubes is obtaine through suitable hand holes in the steel manifolds, these being sealed by steel caps wit copper gaskets.

In addition to their standard convectio superheater, Messrs. Foster have during recen years developed a radiant-heat superheater, fo use more particularly in large furnaces as use for the burning of pulverized fuel, and when th higher superheats are demanded.

This radiant-heat apparatus can be s

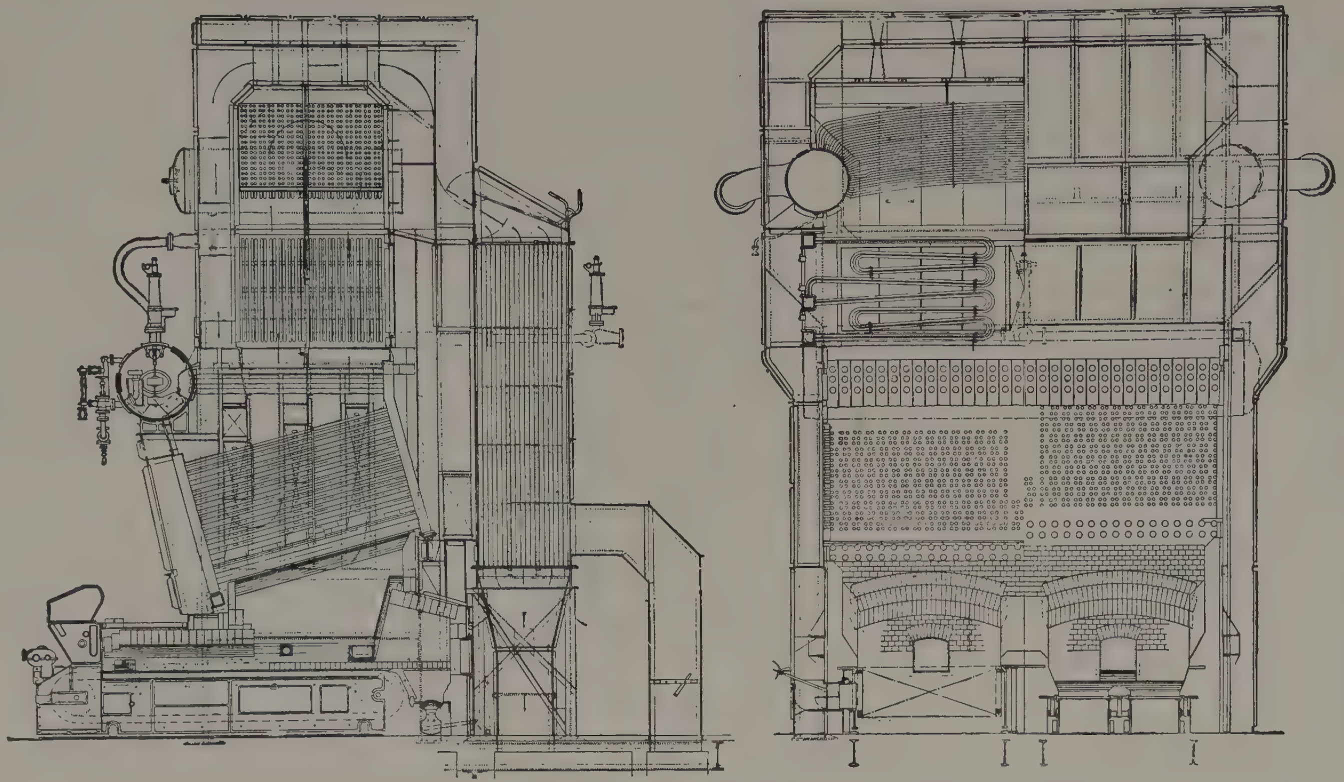


Fig. 36. Water Tube Boiler, Superheater, Steam Reheater, and Air Heater
(Babcock & Wilcox, Ltd)

constructed as to form one or more walls of the furnace, and is made up in rectangular steel sections which fit close together, forming a flat surface, the sections being arranged either horizontally or vertically.

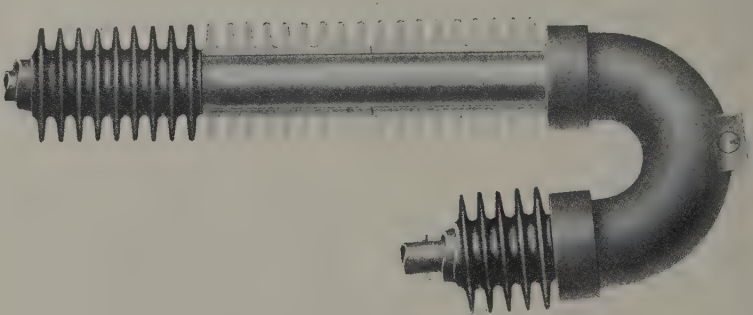


Fig. 37. Foster Superheater Element
Showing cast-iron gills and end connections

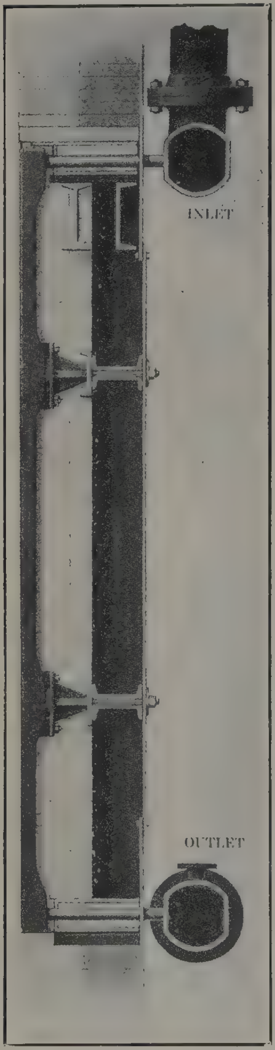


Fig. 39 Typical Vertical Element of Radiant-heat Superheater

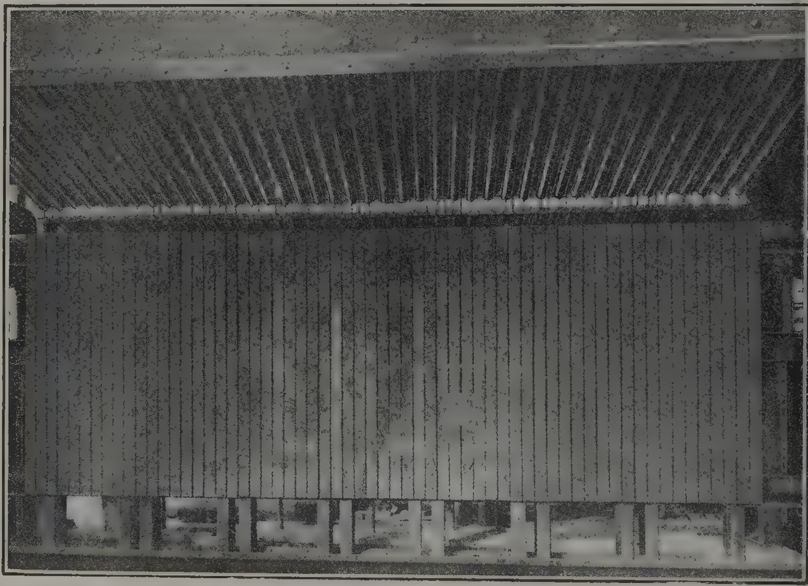


Fig 40. Foster Radiant-heat Superheater Forming Rear Wall of Furnace

Connections are made at each end of the sections, outside the fire zone, to steel manifolds and collecting pipes, the elements being secured through the furnace walls by special bolts to heavy steel bracing girders arranged to provide for expansion, and also to allow any element to be withdrawn when necessary.

To provide against the detrimental effects of soot deposit on the element fronts a special form of soot blower is employed, by means of which the surface can be steam swept as occasion demands.

These radiant-heat superheaters are often run in series with standard convection superheaters, which have already been described.

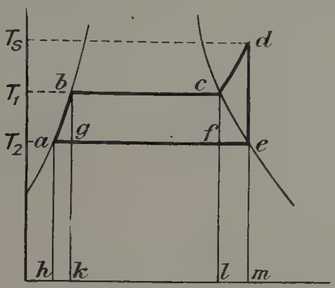
THE THEORY OF HEAT ENGINES

(*Continued from page* 792)

The T-ϕ. diagram is shown. *de* represents the adiabatic expansion, *e* being on the steam line. Then

$$ae = ag + gf + fe$$

i.e. $$\frac{L_2}{T_2} = \log_e \frac{T_1}{T_1} + \frac{L_1}{T_1} + \cdot 5 \log_e \frac{T_s}{T_1}$$



$T_1 = 801°$ F. abs. $L_1/T_1 = 1{\cdot}09$ $\log_e T_1/T_2 = 0{\cdot}269$
$T_2 = 613°$ F. abs. $L_2/T_2 = 1{\cdot}64$

$\therefore\ 0{\cdot}5 \log_e T_s/T_1 = 1{\cdot}64 - 1{\cdot}09 - 0{\cdot}269 = 0{\cdot}281$

$\therefore\ T_s/T_1 = 1{\cdot}754$

$\therefore\ T_s = 1405°$ F.

$\therefore$ superheat $= T_s - T_1 = \underline{604°\text{ F.}}$

$$\begin{aligned}\text{Heat supplied} &= T_1 - T_2 + L_1 + K_p(T_s - T_1)\\ &= 801 - 613 + 873{\cdot}7 + {\cdot}5\,(604)\\ &= \underline{1363{\cdot}7}\text{ B.Th.U./lb.}\end{aligned}$$

Heat rejected $= L_2 = 1007$

$\therefore$ work done/lb. $= \underline{356{\cdot}7}$ B.Th.U.

$$\text{Thermal efficiency} = \frac{356{\cdot}7}{1363{\cdot}7} = \underline{0{\cdot}26}$$

$$1 \text{ horse-power hour} = \frac{33{,}000 \times 60}{778}\text{ B.Th.U.}$$

Each pound of steam performs 356·7 B.Th.U.

$\therefore$ number of pounds of steam per h.p. hour

$$= \frac{\dfrac{33{,}000 \times 60}{778}}{356{\cdot}7} = \underline{7{\cdot}1}$$

FUEL

By Arthur Grounds, B.Sc., A.I.C., Assoc.M.I.Min.E., A.M.C.T.

LESSON VIII

GASEOUS FUELS

Next to solid fuels in order of importance to the British engineer come gaseous fuels, for not only are these fuels used for the purpose of steam raising in steel works and metallurgical plants, but they also find an extensive use in gas engine practice. We have no natural gaseous fuel available in this country, but in America there are large quantities of natural gas used annually. These natural gas supplies consist of mixtures in all proportions of the gases methane, or marsh gas, carbon monoxide, hydrogen, carbon dioxide, ethane, ethylene, oxygen, and nitrogen. For fuller particulars, the reader is referred to almost any of the standard American textbooks dealing with fuels, such as Gebhardt's *Steam Power Plant Engineering*.

Producer Gas. This particular gaseous fuel is probably the most important combustible gas as used for engineering purposes in this country. We have already seen that when carbon in the glowing state is subjected to a blast of air, it burns first of all to form carbon dioxide, whilst if this gas be then passed through a further body of red hot carbon, it combines with the carbon to form the gas called carbon monoxide. This latter gas, which is combustible, forms one of the constituents of producer gas, and combines with half its volume of oxygen to form carbon dioxide, the volume of the latter being the same as the original volume of carbon monoxide burnt.

If, instead of air, we were to blow steam through a glowing bed of fuel, we should find that one of two reactions would take place. If the steam were limited in quantity, compared with the fuel available, the action would be represented by the following equation—

$$C + H_2O = CO + H_2.$$

In other words, 12 lb. of carbon combine with 18 lb. of steam to give 28 lb. of carbon monoxide and 2 lb. of hydrogen. The carbon monoxide produced would occupy the same volume as the steam supplied, *if measured at the same temperature and pressure*, whilst the the hydrogen produced would also occupy the same volume as the original steam under identical conditions. If, however, we were to supply steam in excess, we should find that the reaction would probably take a different course, which could be represented by the equation—

$$C + 2H_2O = CO_2 + 2H_2.$$

In the latter case, the carbon dioxide would occupy half the volume of the original steam, whilst the hydrogen would occupy the same volume as the original steam, all being measured at the same temperature and pressure. In the bituminous gas producer, a mixed blast of air and steam is injected under the grate, so that we have a combination of the above three reactions proceeding simultaneously, and the final gas contains a mixture of carbon monoxide, hydrogen, nitrogen, and undecomposed oxygen, as well as those gases, such as methane and ethane, which are distilled off from the coal purely by the action of the heat in the producer as apart from the action of the steam and air blast. The main difference between the action of air and steam on glowing carbon is that, whereas when air and carbon react heat is evolved, heat is absorbed when steam and carbon react with each other. In the manufacture of water gas, therefore, the mass of fuel in the generator is first of all heated up by blowing the generator with air, much of the fuel being burnt to carbon dioxide, so that a high temperature is attained, and the steam is then admitted so that the fuel is decomposed into carbon monoxide and hydrogen in equal volumes, the gas generated by the action of the steam being diverted along a separate main from that produced during the period of the air blow.

Blast Furnace Gas. This gas is produced in a somewhat similar way, due to the admission of a strong blast of air through the tuyeres of the blast furnace, which air, in its upward passage through the fuel burden in the furnace, generates a considerable quantity of carbon monoxide. The full heat of the fuel is thus not completely utilized in a blast furnace and it is, therefore, imperative, from the point of view of economy, to recover the potential heat in the outgoing gases of the furnace either in gas engines or under steam boilers.

Coke Oven Gas. Where coal is heated in coke ovens for the production of metallurgical coke for the blast furnace industry, large volumes of gas are evolved from the coal in the ovens, and this gas approximates fairly closely in composition to ordinary town's gas. It is, indeed, used frequently for heating and lighting purposes in municipalities where coke oven practice is established. This gas, however, is not the product of the partial combustion of coal, but is a product of the distillation or carbonization of coal.

Below are given analyses of typical examples of various industrial gaseous fuels, together with their calorific values, based on a figure of 68° F., at a pressure of 14·7 lb. per sq. in.

In our next lesson we shall see how these calorific values are calculated, and how the gases are analysed.

Gas	Constituents of Gas Percentage by Volume								Calorific Value	
	CO_2	CO	H_2	CH_4	C_2H_4	CnHm	N_2	O_2	High	Low
Blast furnace gas	11·4	28·6	2·7	0·2	—	—	57·1	—	102	100
Coke oven gas	2·5	6·0	42·0	34·3	2·0	2·0	10·1	1·1	605	546
Town's gas	1·2	6·2	43·9	37·8	5·9	4·2	3·5	0·5	618	558
Water gas (blue)	4·4	44·8	45·6	4·4	0·1	—	0·1	0·5	336	311
Water gas (carburetted)	2·1	24·1	32·4	23·4	12·5	—	3·7	0·5	638	596
Producer gas	5·3	26·1	15·0	0·2	—	—	53·2	0·2	135	127
,, ,,	3·4	25·3	9·2	3·1	0·8	—	58·2	—	155	146

GAS ENGINES AND PRODUCERS

By Arnold Rimmer, B.Eng.

LESSON VIII

STARTING AND GOVERNING

Starting

The gas engine, unlike the steam engine, is not "self-starting," and needs to have the proper sequence of events or "cycle" established through the medium of some external agency.

Small engines may be started by hand, one or more persons pulling rapidly on the flywheel, the operation being assisted by a "half compression" device which causes the exhaust valve to open during the first part of the compression stroke, in addition to its normal opening.

Care should always be exercised in carrying out the above operation, as an element of danger is never entirely absent, and some form of "automatic" starter is a desirable feature on all gas engines and an essential one on most. There are three usual methods by which this may be effected, viz.—

1. The explosion of a charge specially introduced into the cylinder.
2. The use of compressed air.
3. A mechanical barring gear.

Explosion Starters. In the original "self-starters," invented by Clerk, Green, Lanchester, etc., the piston was placed just past the beginning of the working stroke, and gas was admitted to the cylinder and ignited at an orifice in the same. When the cylinder contents were of sufficient richness the gas supply was cut off, the flame struck back and ignited the mixture in the cylinder, thus giving an impulse to the piston. At the same time an automatic valve closed the orifice. A more modern form consists of a small charging pump whereby a mixture of gas and air (or petrol vapour and air) may be introduced into the cylinder and then

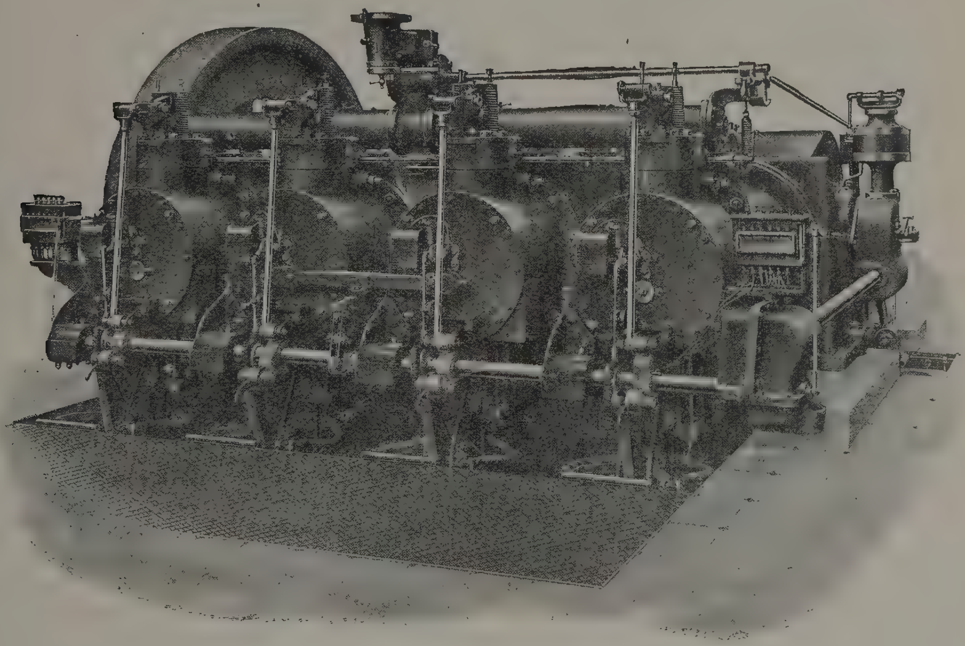


Fig. 38. Arrangement of Compressed Air Starter, etc.
(Premier)

fired by means of a hand tripping device fitted on the magneto.

Compressed Air Starters. This method is reliable and powerful, and is fitted on the majority of modern engines of medium and high power. The air is stored (at pressures varying from 100–300 lb.-sq. in.) in a steel reservoir which is usually charged by a separate compressor driven by a small petrol engine, electric motor, line of shafting, etc. The air is admitted to the engine cylinder through a special valve which is opened for a brief period by a cam. The engine crank is first placed just over the dead centre on the working stroke (in which position the starting valve will be open and all other valves closed), and the air then turned on at the stopcock. In the "Premier" multi-cylinder engine a cam-operated distributing valve allows air to be admitted at the correct moments to the two centre cylinders. It enters through non-return valves, so that as soon as explosions begin the admission of air automatically ceases, even if the operating mechanism is still in gear (Fig. 38).

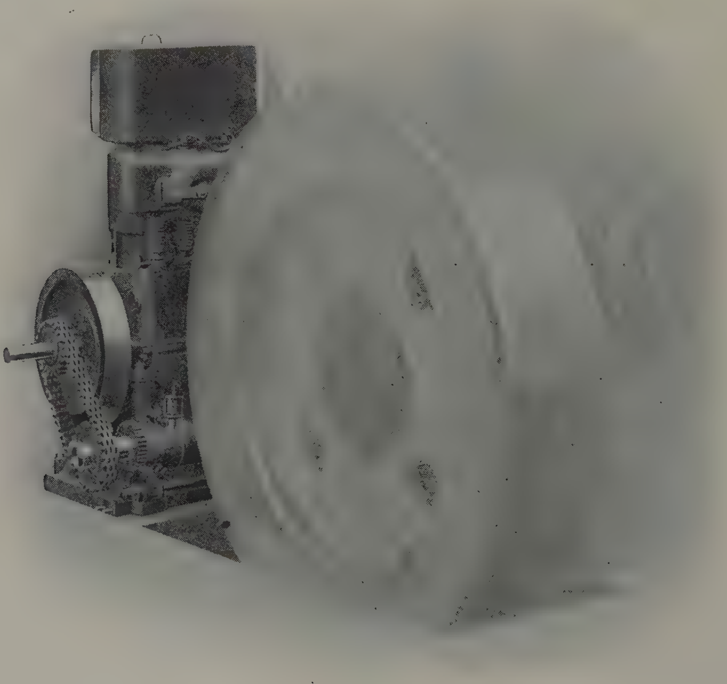


Fig. 39. Friction Starting Gear (National)

Mechanical Starters. In this system th
engine crankshaft is rotated by an externa
source of power at a speed which, though com
paratively slow, is yet sufficient to cause th
mixture to be drawn into the cylinder an
ignited in the ordinary manner, thus estab
lishing the cycle of events. Thus, a pinion
provided with an internal "free-wheel" may
be capable of engagement with teeth on th
inside of the fly-wheel rim. Normally, th
pinion drives the fly-wheel, but when ignitio
commences and the speed of the engine suddenl
increases, the fly-wheel "overruns the gear,
and the pinion is then disengaged from t
fly-wheel.

The arrangement supplied by the Nation
Gas Engine Co., Ltd., with certain of the
vertical engines (90–270 h.p.). is shown
Fig. 39.

It consists of a two-cylinder petrol engi
driving a friction pinion of "Ferodo" materi
which may be brought in contact with t
smooth outer surface of the fly-wheel. This
effected by mounting the pinion shaft

eccentric bushes, which can be oscillated by means of a foot lever placed in a position convenient for starting.

GOVERNING

The actual "governor" on a gas engine is usually of the centrifugal type, driven direct or through gearing from the crankshaft. (See Fig. 40, and also Figs. 20 and 24, pages 465 and 467.)

Alterations in load and, therefore, speed, cause a radial movement of the governor balls, and consideration will here be given only to the methods of utilizing this movement to adjust the supply of gas and air to suit any particular load, while maintaining the engine speed sensibly constant.

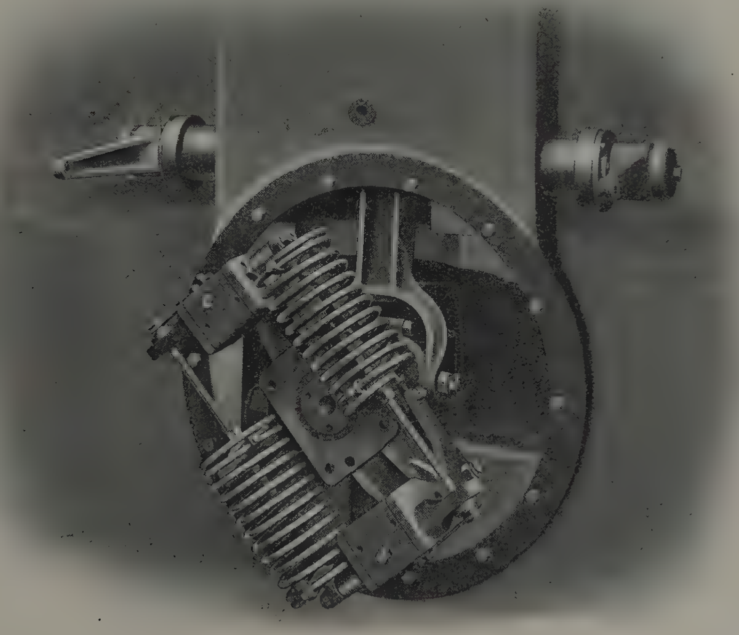


FIG. 40. GOVERNOR (NATIONAL VERTICAL TANDEM ENGINE)

The essential features of the two main systems are shown in the following table—

System	Proportion of Gas to Air	Volume of Charge	Compression Pressure	Pumping Losses
Qualitative	Varied	Constant	Constant	Constant
Quantitative	Constant	Varied	Varied	Varied

Hit and Miss Governing. Reference has previously been made to this system, and it may be regarded as a special case of qualitative governing. The principle is that a full charge of gas is always admitted at normal speed, but that it is cut out entirely above a certain limit of increased speed. This means that air only is then admitted, the cylinder is well cooled and scavenged, and the next explosion is more violent. The fluctuations of speed thus set up are fairly large and heavy fly-wheels are required. As fitted on a 50 h.p. horizontal National engine this system is ilustrated in Fig. 41.

The gas valve is operated by a knife-edged "pecker" engaging with a V-groove in a small block which, in turn, presses against the end of the gas valve spindle. The block may be moved by the governor, in which case the pecker cannot engage with it and the gas valve remains closed. The working parts are light and the required movement comparatively small, and so the actual governor does not need to be very large or powerful.

This system is very economical as regards gas consumption, but it is not suitable for large engines owing to the uneven turning moment produced, and also because "knife edge" opening of large valves is not satisfactory from a mechanical point of view.

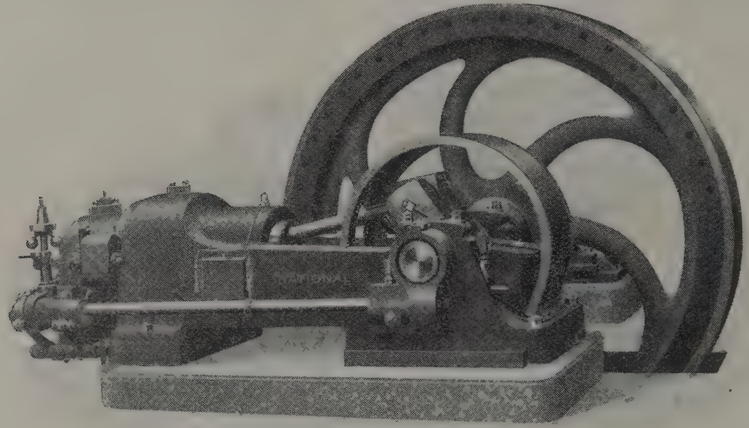


Fig. 41. 50 H.P. National Engine

Variable Gas Admission. In this type of qualitative governing the time of opening of the gas valve is varied (the closing always taking place at the end of the suction stroke), and the air admission remains constant. This gives explosions of varying intensity, but since the compression pressure is constant, the method is very suitable for large engines with heavy reciprocating parts. At light loads there is a possibility of the mixture being too weak to burn, though explosion is generally assured owing to the fact that the majority of the gas will be near to the ignition plug. Fig. 42 illustrates, in diagrammatic form, a governing device of this nature as fitted to 500 h.p. tandem engines at one time manufactured by Messrs. Crossley Bros.

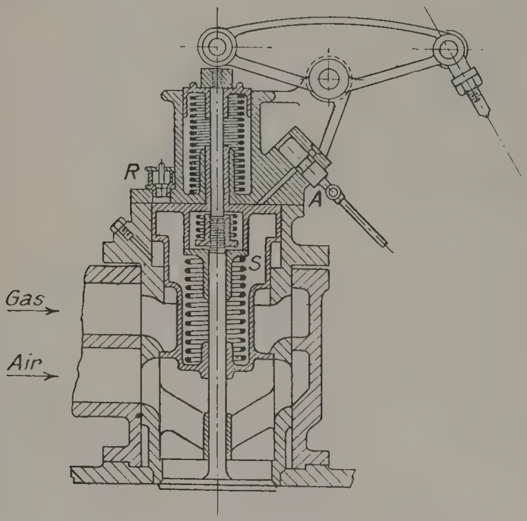


Fig. 42. Variable Gas Admission Governing (Crossley)

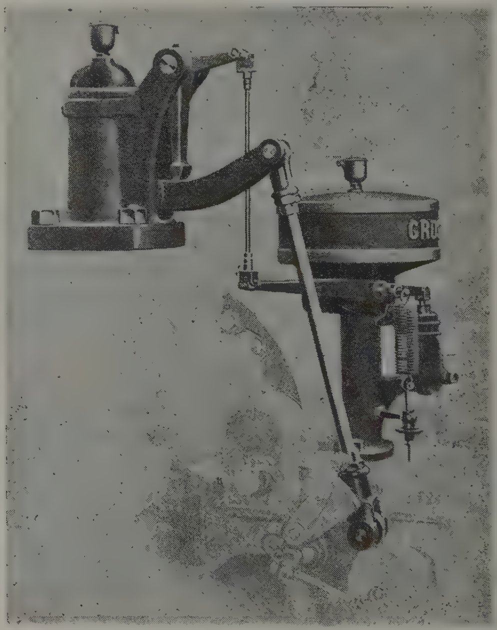


Fig. 43. Variable Admission Governing (Crossley)

The gas valve is mounted on the admission valve spindle and is prolonged to form an annular piston working in an air-tight cylinder, the only connection to the atmosphere being via a small valve *A*, which is under the control of the governor. The larger the opening through this valve the faster will the gas valve follow the admission valve (by the action of the spring *S*), and the richer will be the resulting mixture. A converse action takes place as the governor closes the valve *A*, when the speed tends to rise. The gas valve is closed positively when the admission valve returns to its seat by means of a collar on the admission valve spindle, any air which may have leaked into the vacuum cylinder being expelled through the relief valve *R*.

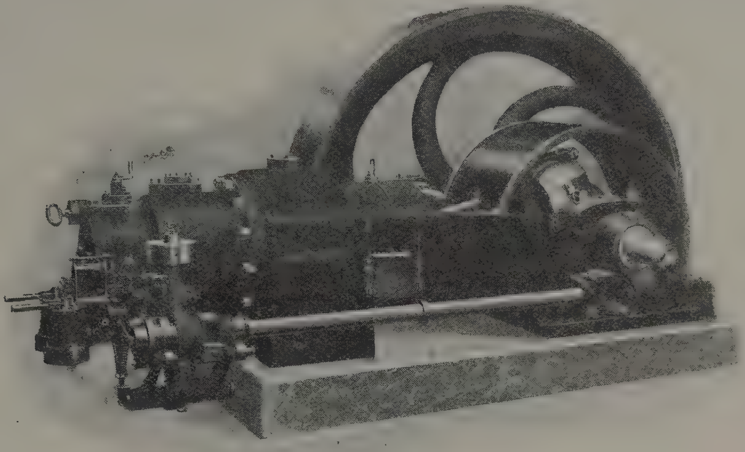


FIG. 44. 95 H.P. NATIONAL ENGINE

Another method of variable gas admission is that of throttling the same in the supply pipe, but allowing it to flow into the cylinder throughout the stroke, the quantity of air admitted remaining constant as before. This is the method used on the "Premier" multi-cylinder engines described on page 467. (See Fig. 24.)

Quantitative Governing. This system is usually applied in one of two ways, viz.—

(*a*) Throttling.
(*b*) Variable lift to inlet valve.

Throttling may be that of the mixture before passing the inlet valve, or the governor may operate separate valves in the gas and air supply. This is the system fitted on the National Vertical Tandem engine, and also the Galloway engine described on page 587. In some cases, the first movement of the governor is arranged to affect the gas supply only and then, when the mixture has been somewhat weakened, both gas and air throttles are operated together.

The system of varying the lift of the inlet valve is a very usual one, particularly on larger engines. The arrangement as fitted on the Crossley engines is illustrated in Fig. 43. (See also Fig. 23, on page 466.)

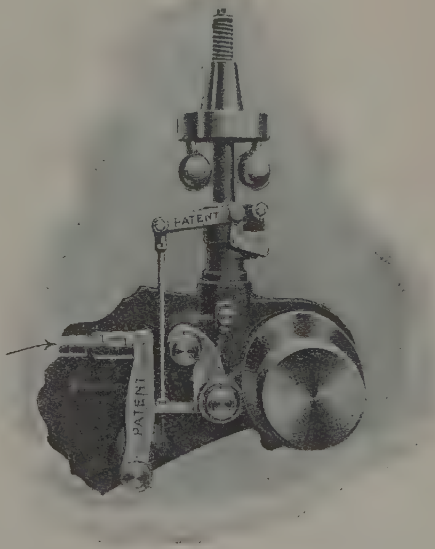


FIG. 45. VARIABLE ADMISSION GOVERNING (NATIONAL)

It consists of a movable fulcrum arm operated through linkwork by the governor. As the speed rises the fulcrum arm moves inwards and so the lift of the inlet valve is reduced, with a converse action when an increasing load tends to lessen the engine speed.

On certain of the National Horizontal engines the same result is obtained by means of a plate whose position between two operating levers is directly under the control of the governor (Figs. 44 and 45).

Quantitative governing gives efficient combustion at all loads, but owing to the variable compression pressure caused by the system, the inertia of the reciprocating parts is not always cushioned on the compression stroke. At very light loads a partial vacuum may be created in the cylinder during the suction stroke, necessitating very strong springs to prevent the valves being lifted off their seats by atmospheric pressure, unless they are operated in both directions by double-faced cams as in the 2,000 h.p. Premier engine referred to on page 585. (See Fig. 32.)

In some cases provision is made for running at light loads by combining the hit and miss systems with either variable gas admission or quantitative governing.

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc. (Vict.), M.I.A.E.

LESSON I

HISTORIC SURVEY OF DESIGN

The history of automobile engineering gives us many clues to the present-day design of automobile vehicles.

The automobile on roads came into being at the same period as the automobile on rails. Owing to the crude materials available and the lack of precision machine tools the weight was excessive for the power developed. The poor roads would not carry the weights, and a more rapid development took place on the railway, where the track was more easily made to carry the weight. The road automobile was killed by legislation imposing severe speed restrictions, leading to the development of heavy traction engines which could pull paying loads at the speeds allowed. In India and France efforts were made to use fast vehicles of this class, but tyres, springs, and roads were not equal to the work. The pneumatic tyre was designed in 1845, but until 1889 was neglected.

The pedal cycle was developed in the last quarter of the nineteenth century, and efforts towards mechanical propulsion were many. Electric and steam vehicles were tried, but the internal combustion engine gave a new direction to research, and in the early eighties tricycles were made with internal combustion engines. At the same time the steam tramway and steam launch had given opportunity for research with light weight boilers and engines. The nineties opened with the development of steam lorries and omnibuses (largely designed and manufactured by launch builders), and light weight cars developed from the bicycle.

The rapid development of the petroleum spirit engine for launches brought this engine into commercial production in sizes large enough for four-seated carriages.

The launch engine was not restricted in height and had almost universally two cylinders. Carriages at this time—1890–1895—were of every form: horizontal cylinders with chain drive, vertical cylinders in front with chain drive engines under front seat, under rear seat, vertical crankshafts, and radial engines with rotating cylinders were all tried. The most reliable vehicles in racing and touring were, however, those with vertical two-cylinder launch engines in front with longitudinal shaft, a bevel cross shaft with differential, and chain drives to the rear wheels. This was not ue to any inherent value in the type, but these ngines had been extensively commercialized n the hard work of motor boats and hence he makers of cars could buy them with confi-ence at a reasonable price.

In order to enable hills to be climbed without sing too low a gear reduction on the level, everal gear reductions were fitted usable in urn, similar to machine tool practice. A riction clutch enabled these to be engaged at vill or the engine to run free.

This was a much better mechanical job than elts on stepped pulleys, expanding pulleys, riction drives, expanding chain sprockets, ariable stroke levers operating ratchets or ther methods then being tried, and, except or certain special cases mentioned later, is till the standard practice. It is simple, nechanically sound, and not easy to put out f order.

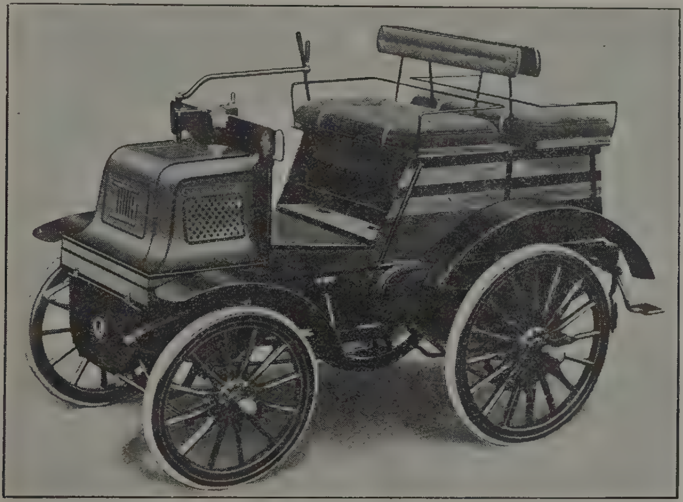


Fig. 1. Daimler Dog-cart of 1896

Ignition up to this time had been by a plati-um tube kept hot by a lamp. The lighter ehicles used an electric spark, but the apparatus sed was very crude, and the tube was a more eliable method until specialists in ignition pparatus appeared. The magneto appeared in 900.

Cooling water was usually boiled away by he engine and replenished as is done in small gricultural engines to-day. Circulation through radiator was then adopted. At first the cooler vas under the car, where it soon became coated vith mud and became inefficient. The forward position succeeded and the general outline of the modern car was standardized by public taste. Features of successful racers were adopted whether useful or not, and by 1902 the general arrangement was settled.

Up to November, 1896, the legislation which had killed the automobile in Britain still remained in force, 4 miles per hour in the country, 2 miles in town. A man had to walk in front carrying a red flag. In the above month Parliament permitted vehicles not over 2 tons in weight to proceed at 12 miles per hour, up to 3 tons 5 miles per hour. The steam vehicle had developed on separate lines, and although many of them weighed over 5 tons they were all weight marked 2 tons 19 cwt. 3 qr., as there was no power given to the authorities to weigh them. This Act was amended by the law of 1903, which is still in force—20 miles per hour under 2 tons, 12 miles per hour over 2 tons on rubber or soft tyres, 5 miles per hour on steel tyres.

An enormous impetus was given to the movement by these concessions and development proceeded on many lines.

The private car developed in four classes—

1. The large luxurious car.
2. The good four-seated family car.
3. The small light cheap car.
4. The sporting car.

These, of course, overlap each other, but the designer's problems are substantially different.

1. Comfort, appearance, and speed required,

First cost and operation cost are not of importance, but reliability is an essential.

2. First cost must not be high. Appearance must be good and like the bigger car. (This has standardized appearances more than is commonly thought.) Is sold in quantities and can be designed for quantity production.

3. Low price, accommodation usually scant, and appearance need not closely resemble the big car. Speed is usually required even at expense of endurance.

4. The sporting car may be a design of any one of above classes, improved above its fellows.

Owing to the proximity of the countries, automobile design in Britain, France, Germany, Belgium, and Italy developed along similar lines. French chauffeurs in England, English mechanics in France, helped in this uniformity, but the United States of America developed its designs on its own lines.

In the first place they had no network of roads. The railways had the inter-urban traffic before roads came, and the crossing of the continent by car was a great triumph partly done on railway tracks.

This developed the high car. The single-cylinder horizontal petrol engine was early developed in America as a general purpose engine, sawing logs and running machinery, and was adopted by many early makers of automobiles, just as the launch engine was in France and Germany.

There not being the same social traditions in America the carriage was not developed. For many years price governed the market, and foreign cars were imported by wealthy people. The same conditions still hold in some colonial markets. In later years America has followed and now often leads European practice. The petrol engine appeared in goods vehicles about 1900, but for 3-ton loads steam still held its ground for some years, though now only used for loads of 6 tons and over.

The large passenger vehicle had to contend with serious legal difficulties owing to weight restrictions until 1904, when the very rapid development of motor omnibuses commenced The legal restrictions in London have produced a type of vehicle which gives the maximum passenger capacity for a given weight, length and width.

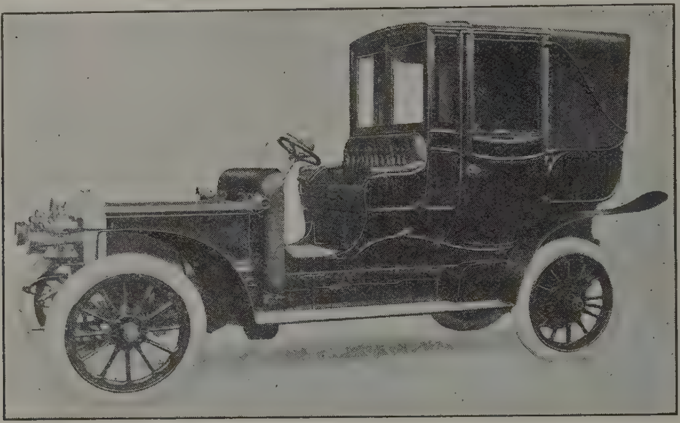


Fig. 2. Daimler Landaulet of 1906

The introduction of the pneumatic tyre on heavy vehicles began just prior to the Great War, and since the Armistice has enormously increased the field of the motor omnibus and public coach.

Steam has been tried many times for private and public passenger work. It is an attractive proposition with its easy running and silence but it is noticeable that the only steam vehicle in use are lorries devoid of any automatic feed for either fuel, water, or air, and none can run thirty miles on one filling of water.

A high degree of technical skill is necessary to drive a steam car and, in 1904, the cheapes cars were American steamers. Although high technical skill is not always a well-paid possession, it does not hold vice versa that by paying low wages one can obtain a driver with high technical skill.

Electric Vehicles. Many efforts have been made to develop these, but there is a well-defined field of service for them where frequent stops are made and roads are not very steep, such as for refuse removal. Little skill is required in driving. The weight is high and batteries do not run more than 35 miles on a charge.

The fundamental difficulty is that the materials required for an electric motor and storage battery have to be chosen for their high or low electrical resistance, high or low magnetic properties or chemical reactions in case of batteries, and copper, rubber, mica, soft iron, paper, and lead are not strong materials.

The survival of the four-cylinder and six-cylinder vertical engine in the front of a car, driving by a plate or cone clutch through three- or four-speed sliding-gear type of box with "direct" drive on top, a worm or bevel driven live axle and four pneumatic tyred wheels, a body with two seats facing forward entered by side doors, steered by a hand-wheel on an inclined pillar, and controlled by a clutch pedal under the left foot and brake pedal under the right foot, is very much like the survival of the present type of locomotive. It happened, it worked, buyers chose the type they fancied, others copied successful sellers.

An examination of old technical papers, 1895–1905, shows ten years of intense study, handicapped by inadequate materials, restrictive laws, intense opposition of other vested interests. The curious thing is that the first application of the petrol engine to locomotion was to a motor-cycle, and up to quite recent times the motor-cycle was just as crude in design, deficient in power, reliability, comfort, and cleanliness as the motor-car of 1901. As recently as 1914, pedalling gears were fitted to leading makes.

Great strides have been taken and the stigma is removed to a large extent. Such is, in brief, a history of the rise of a great industry, a great social factor, a mechanical training for a very large section of the community. Those chiefly associated with its rise are still watching its development.

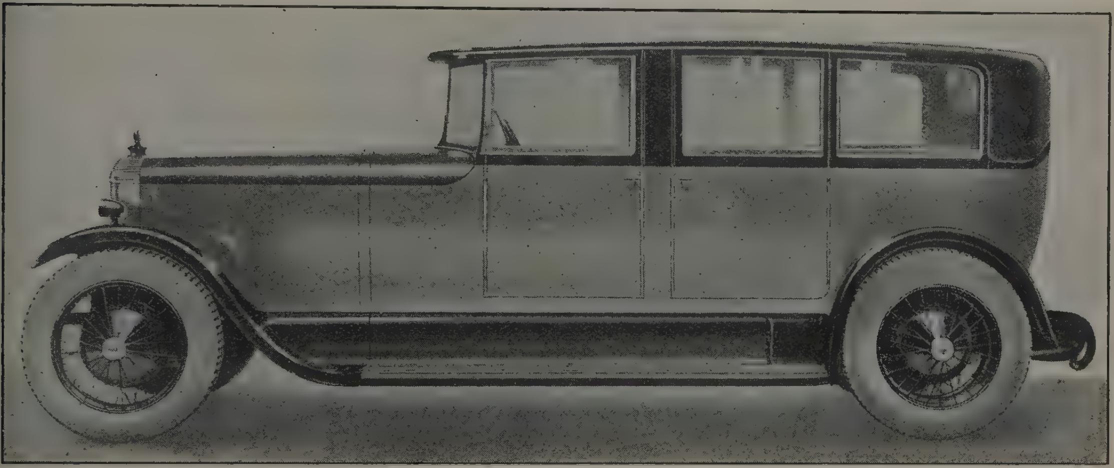


FIG. 3. DAIMLER ENCLOSED LANDAULET MODEL P.S.1

FITTING AND ERECTING

By Joseph G. Horner, A.M.I.Mech.E.

LESSON IX

CRANE WORK

The crane making industry is a very extensive one. It ranges from the lightest hoists and crabs to the most massive dock and harbour equipments. It includes second-class work put into contractors' cheap cranes, and that for the best class of overhead electric travellers and allied forms. Hand, steam, and electric power are all employed. Speeds are slow or rapid, loads may range from half a ton to 200 tons. Cranes are fixed or portable, they travel on rails on the ground or on overhead gantries. They are built single or in sets. Their construction embraces all the methods of the machine and fitting shops. They enlist the services of the skilled fitter and erector, for, except in the preparation of some standardized parts and units, the interchangeable system is not readily adaptable to crane building, because of the very limited numbers of one class and size that are required. Even in the smaller cranes of a given class and of the same power, modifications in some details are often introduced to comply with specifications.

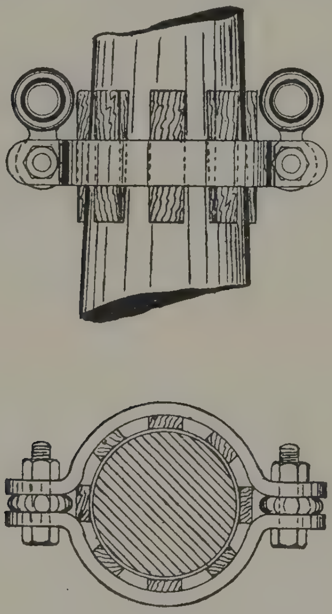


Fig. 55. Slinging Crane Post

Fixed Cranes. Most cranes, the term applied to hoisting machines with jibs, consist of a base portion, or foundation, and a superstructure. These are prepared separately, and brought together on completion, the jib being added afterwards. In these cranes the post is a fixture, on which the superstructure is suspended and around which it is slewed with rollers running on a belt turned on the post, or on a path on the foundation plate. The post is stepped into a foot several feet below the ground level, passing through the base plate on the level, which is connected securely to the foot, or to a foundation plate, or anchored in concrete with bolts and broad washers. In the shops, wells are provided, masonry-lined over, in which the plates and posts are erected temporarily.

To fix a post in the bore of its base plate, it is slung in the overhead crane, and entered in the bore, when, being struck all round with a sledge, or a heavy wooden mallet, while the slings are slackened, it will descend gradually into its place. It is embraced with clips (Fig. 55), having lugs to receive eye-bolts for the sling chains, or ropes, and with strips of wood interposed to prevent slipping of the clips on the tapered post. When a post has an enlarged collar at the top this can be embraced with the sling directly. Hempen ropes are generally to be preferred to chains for slinging moderate loads. If kinks occur in chains a few slight taps with a hammer will straighten them out, and prevent sudden jerks in suspension. The friction of a post in its hole is generally sufficient to hold it from rotation, but if the fit is slack a set screw may be inserted through the boss. On the post the superstructure, completed elsewhere, is set bodily with the overhead traveller.

Portable and Travelling Cranes. These have their superstructures supported on posts that are stepped into trucks mounted on wheels to run on rails. The breakdown cranes for railway service have to conform to the standard specifications of the companies in respect of wheels, axle boxes, horn blocks, springs, and brakes, but those for common use have plain bearings of the plummer block type, and plain wheels of cast iron, chilled on the treads. The heavy cranes are modified to carry high loads, with spring bearings, and rolled steel tyres shrunk on centres of cast iron or cast steel. The travelling gear is usually driven from the superstructure from a vertical shaft, bevel wheels, pitch chains, and sprocket wheels; except in some of the heaviest cranes where engines are attached to the trucks for travelling only, driving directly to an axle, and have link reversing motion.

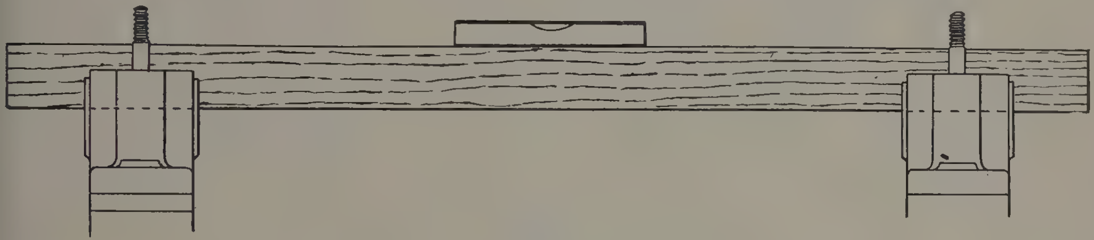


FIG. 56. LEVELLING BEARINGS

In all except the heaviest cranes the truck is turned upside down for the fitting of the bearings, axles, and wheels. The bed is levelled up on timber blocking kept for general use—pieces of balk, batten, quartering, with wedges, set with long straight-edges and spirit-level. Bearings cast or bolted on are tested by their concavities with a straight-edge and level (Fig. 56), and all other parts are aligned from these bearings, parallel or at right angles. The smaller cranes have trucks of cast iron. But in others, and in the best, a built up steel frame receives a casting in the centre for the post, and the ring gear for slewing. In any case the tooling is all done before the work comes into the hands of the erector.

It is necessary first to check the plated frame to ascertain whether it is truly rectangular, and not winding. If inaccurate it will be returned to the platers for correction. The centre casting is fitted, and the whole turned over bodily and levelled on blocking, the various bearings are set by centres, the bolt holes drilled and reamed through them and the plating, and secured. The axles are run round in place before the final bolting up, and the bearings eased with a file and scrape, if necessary, until the axles will turn easily. There should be a little clearance at the sides. A typical bed at this stage, turned over with the wheels on the track is seen in Fig. 57, ready for the insertion of the post, to be slung and inserted as just described for the fixed cranes.

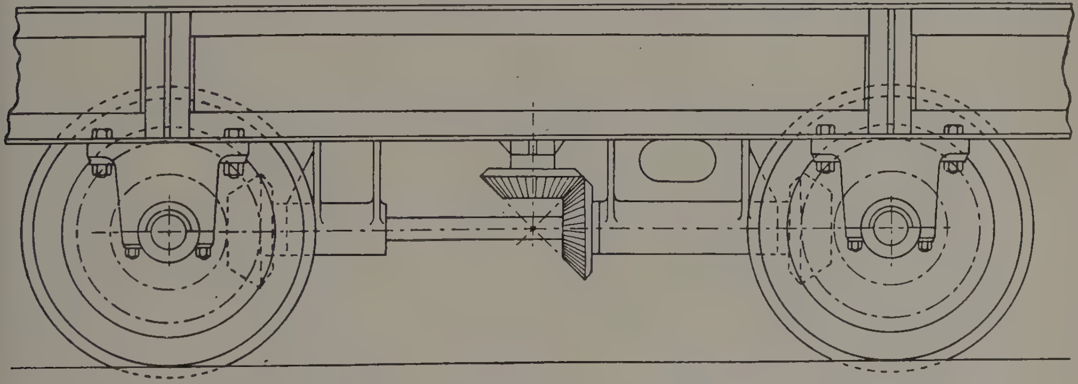


FIG. 57. CRANE TRUCK COMPLETED FOR THE SUPERSTRUCTURE

The work on the superstructure much exceeds in quantity that on the truck. The fitter-erector or a mate prepares it with helpers, who take separate sections of the task. It comprises two side frames or cheeks connected with cross-girders. The frames carry the engines and the shaft bearings. The cross girders surround the post and receive various brackets, roller bearings and shaft bearings, diversified with the type of crane. These are of cast iron, except in the largest cranes, where the frames and often the girders are built up with steel plate and angles. In these the bearings are of cast iron fitted to the plating, and this work, with the marking out, devolves on the fitter. The pattern-maker tries the patterns in place with chipping strips, but the fitter always has to make some corrections in the castings. Two types of these are shown in Figs. 58 and 59. Undivided bearings are not used in good cranes, because of the difficulty they cause in the insertion and withdrawal of the shafts endwise. They are always divided, and fitted with bearings of gun-metal, which can be replaced when worn. Fig. 58 is for a drum shaft, Fig. 59 for an engine shaft. Circular brasses, one with a stud to prevent rotation and a cast-iron cap, are fitted in the first, and square brasses with a steel cap in the second. The outward pull on the drum-shaft is resisted with through bolts, the stress on the engine shaft being mostly vertical, studs suffice. In each case lock nuts are necessary. The outlines of the chipping strips, and the bolting to the frame are seen.

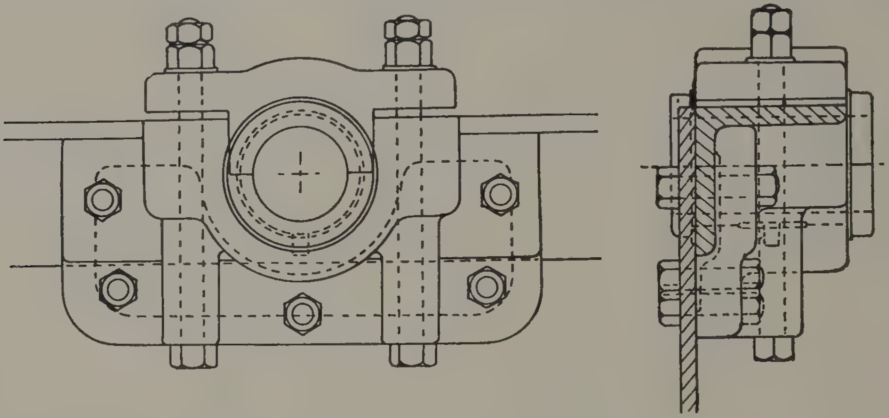


Fig. 58. Fitting Cast Bearings to Steel Plating

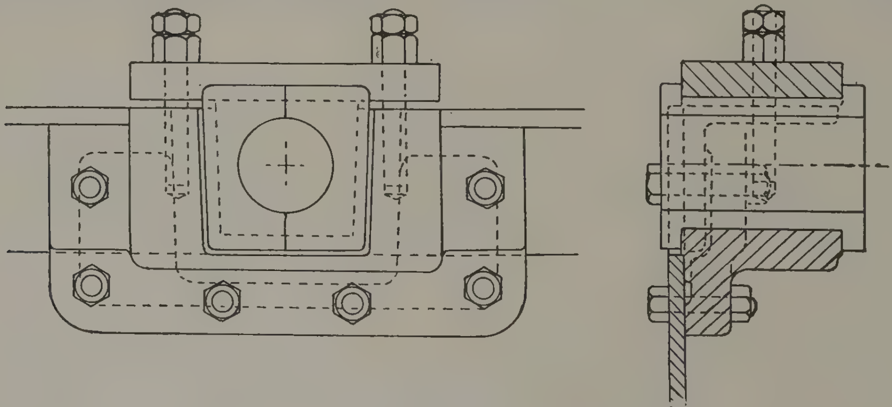


Fig. 59. Fitting Cast Bearings to Steel Plating

There are many related parts that have to occupy very precise positions. The more important are the centres of the gear wheels, and the relation of the engine shaft to the cylinders. These have to be determined before the bearings are bored, or, if this has been done the drilling and reaming of the bolt holes in the plate must be left to the last. Gears are only cut in the high-class and electric cranes, in others the teeth are cast from patterns, and tooth profiles, and diameters are not to be relied on for close dimensioning. The centres then have to be accommodated to the meshing of

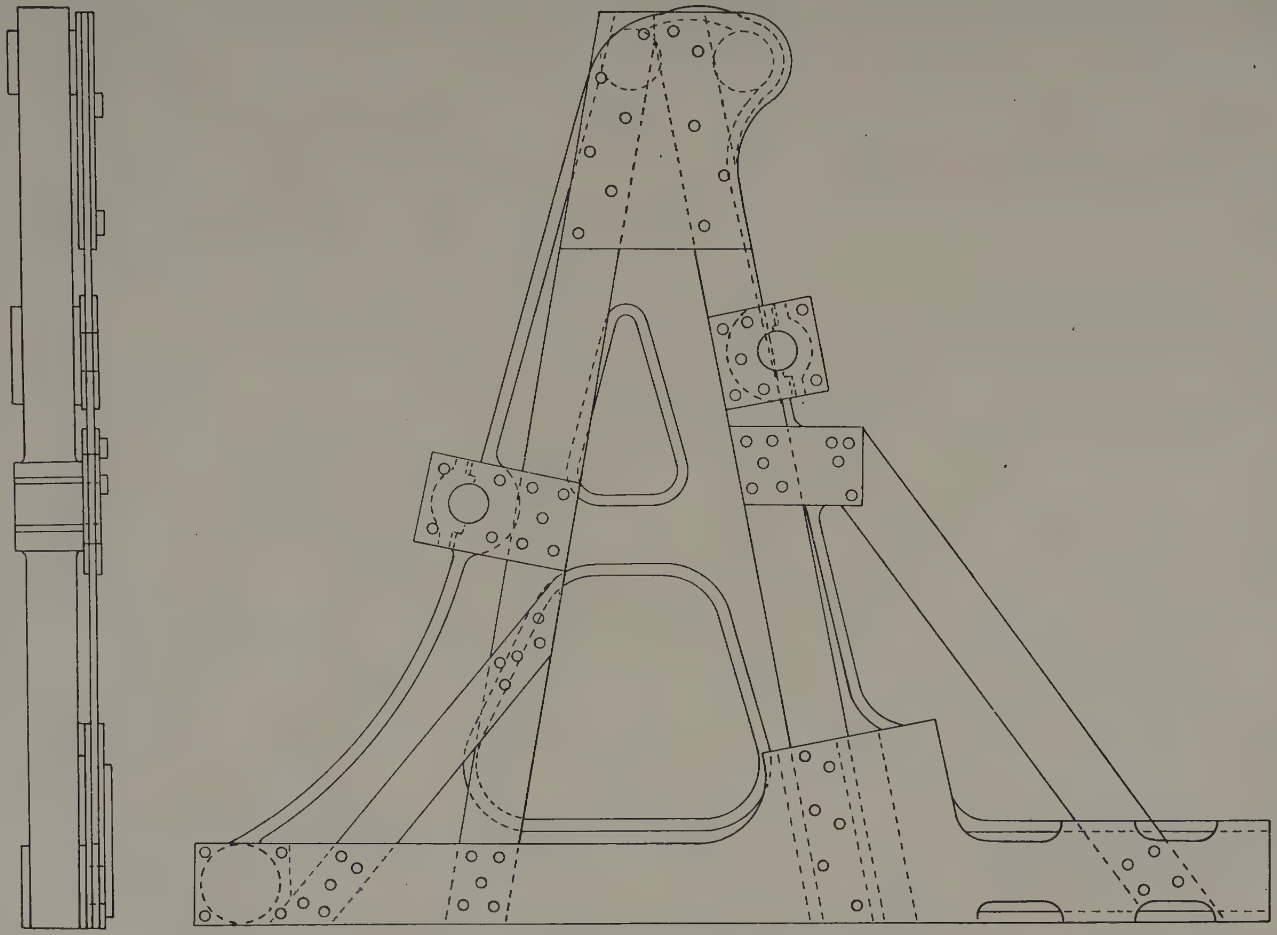


Fig. 60. Templet for Lining-off Crane Cheek

the wheels, of which there are generally not less than three, and often four in mutual relations, on the engine shaft, the drum and intermediate shafts. The gears are set in correct engagement, the bores bridged, and the centres taken with trammels, and transferred to the bearings, also bridged, whence the bores are described. If these should be marked directly from the drawing, without reference to the castings, some teeth would be running too freely, with backlash, and others would jamb with risk of fracture, and require tedious easing with files. When the smaller cranes are in a measure standardized, a templet may be used to mark out the centres and bores of bearings and bolt holes. Such a one is shown in Fig. 60 laid on a cast-iron cheek. It is framed of thin sheet iron, and is light enough to be handled.

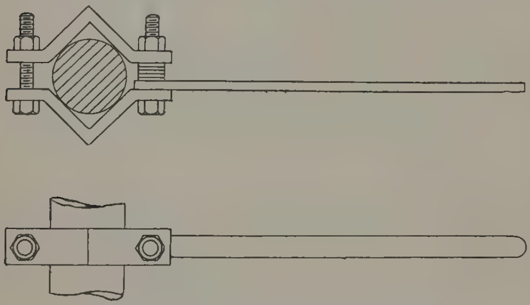


FIG. 61. DEVICE FOR ROTATING SHAFT IN ITS BEARINGS

Two methods of boring and facing are employed. One treats each frame separately under a vertical spindle machine, the other bores them together after the cross stretchers have been bolted in their places, using a bar and cutters in a horizontal spindle machine. The latter is to be preferred, as the holes are more likely to be in alignment. The boring and facing have to be done in two stages, that for the seatings first, and the brasses subsequently.

When these are completed, the frames connected with the cross girders are levelled on wood blocking, the datum being a shaft inserted in the engine shaft bearings and, in the other direction, the vertical height from that to the drum shaft. The engine cylinders, which have been machined and completed, are fitted first. These may stand with their axes vertically or diagonally, or sometimes horizontally. A templet of wood or sheet metal is cut to fit on the face of the upper flange of the cylinder and to enter its bore. A centre line on the templet reaches to the centre of the engine shaft, and the cylinder, held temporarily with clamps, is adjusted until it is aligned with the shaft, when the bolt holes are drilled and reamed through its feet and the frame, and it is bolted permanently in place.

There is much detail in connection with the engine fittings. The eccentrics will have been keyed on the shaft, with all the gears and clutches which it carries, before the shaft is inserted. The link motion follows, the relations of the link and rods to the slide valve are obtained from the valves, which are set for lap and lead, the steam chest cover being removed for the purpose. The connecting rods from the piston-rods are fitted to the pins of disc cranks on opposite sides of the frames, the crank pins being at right angles.

The various shafts may not work freely in their bearings consequent on a lack of strict alignment. If this happens, the shaft journals are smeared with a thin film of red lead, and the shaft is moved round small arcs with a lever (Fig. 61) to show where contact occurs, and the bearings are scraped until the fit is easy. Too tight a contact would result in friction and heating up when in continuous work.

After numerous levers for changing gears and operating clutches have been fitted, and friction rollers and minor details attended to, the superstructure is lifted over the post. The boiler is set in place on the foot, and pipe connections are made. Finally, the jib, prepared complete, is hoisted into place and united to the frames. The jib is prepared in the plating shop, but the fitter has to attach feet, pulleys, bearings, tie rods, snatch blocks, and reeve chains, wire ropes or luffing gear.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON IV

WROUGHT IRON

Cast iron has but little tenacity and neither malleability nor ductility. On the contrary, wrought iron, which is the purest commercial form of iron, is extremely malleable and ductile, fairly tenacious and easily weldable at suitable temperatures.

The methods employed for the production of wrought iron may be considered as falling under two heads—

1. The direct reduction of the iron from the *ore*.
2. The purification of *pig iron*.

Depending on the simple chemical reaction—

$$\underset{\substack{\text{Iron (ferric)}\\\text{oxide}}}{Fe_2O_3} + \underset{\text{carbon}}{3C} = \underset{\text{iron}}{Fe_2} + \underset{\substack{\text{carbon}\\\text{monoxide}}}{3CO}$$

the direct method, used from the earliest days of history, differed in the details of its application according to the advancement of the peoples. All heated the ore in contact with a carbonaceous fuel, usually charcoal, in a current of air. The small yield of metallic iron in comparison with the weight of materials originally charged, the time taken and the high fuel consumption, all operated against the commercial success of the principle, even despite the many attempts during the nineteenth century, in Germany and elsewhere, to bring the then modern methods to the assistance of ancient experience.

Direct processes are still used in some of the less civilized parts of the globe and, to some extent, in a few districts where local conditions conspire to furnish a field of rich, easily mined ore and an adequate supply of fuel. Their contribution to the world's output, however, is so small that it can be said that the demands of industry are almost entirely satisfied by the product of indirect processes.

Iron, as produced in the blast furnace, contains roughly from 5 to 10 per cent of impurities and possesses neither malleability nor ductility. To give it these essential qualities it is necessary to remove the impurities. This is achieved by *controlled oxidation*, either by heating the pig iron and fuel together in a stream of air (the open hearth), or in the reverberatory furnace, wherein the fuel and pig iron are in separate, intercommunicating chambers, the oxygen being supplied by means of added oxidizing materials.

Two open hearth processes are worked in Sweden, the Walloon and the Lancashire hearth, Swedish white pig—an iron of exceptional purity—being the basis of both.

The Walloon Hearth. In the Walloon process charcoal is charged in a rectangular chamber into which cold blast is introduced by means of a single tuyere. Large D shaped pigs, which, in some cases, weigh as much as 15 cwt. each, are propelled on wooden rollers, by hand power, into the charcoal fire and subsequently melt. The molten globules of iron drop down through the heated carbon and, as more pig iron is brought forward, collect into a pasty mass on the hearth beneath. In so doing, incomplete oxidation of the carbon, silicon, and manganese occurs and, to complete the reaction, the leading man, with a rod, works up the pasty iron more and more into the blast and thus exposes it still further to the oxidizing influence. When the operation is adjudged complete the iron, from which practically all the carbon, silicon, manganese and some of the sulphur and phosphorus have been removed, is in the form of a sponge, in which little globules of slag are interspersed. This intermingled slag must obviously be expelled. The iron sponge, usually weighing about 70 lb., is therefore taken to a water helve under which it is submitted to pressure, shaped, and finally hammered out into bars of about 3 in. by ½ in.

The Swedish Lancashire Hearth. For the Lancashire hearth the iron from the blast furnace is cast into flat, rectangular cakes and the process, though based on a similar principle to the Walloon, is worked on a much bigger scale and by hot blast. Two or three tuyeres are used and the waste gases are utilized for a preliminary heating of the iron cakes. These are afterwards drawn down into the furnace, melted and worked as before described. The sponges, which may weigh as much as 200 lb. each, are then conveyed to a shingling hammer for expulsion of the slag, and ultimately finished under the rolls, into bars of similar dimensions to the foregoing.

The Effects of Refining the Iron. The effects of this controlled oxidation are best demonstrated by the following comparative tables which give the chemical analyses and tensile tests of (1) the Swedish white pig, and (2) the refined iron.

	(1) Per cent	(2) Per cent
Combined carbon	3·50	0·05
Free ,,	·50	—
Silicon	·25	0·03
Manganese	·30	·10
Sulphur	·02	·01
Phosphorus	·03	·02
Iron (by difference)	95·40	99·79

	(1)	(2)
Elastic limit	11	11 tons sq. in.
Maximum stress	11	21 ,, ,,
Elongation	—	40 per cent on 2 in.
Reduction of area	—	70 per cent on ¼ sq. in.

The South Wales Process. The high sulphur contents of British white pig iron render it quite unsuitable for the production of wrought iron by either of the processes previously mentioned and, therefore, to overcome this difficulty, the South Wales process was devised. In this, grey iron was taken as a basis and treated in two operations. In a hearth called the "refinery" the grey pig was, by remelting in an oxidizing atmosphere, changed to white iron, the carbon being thus in the combined form instead of its previous graphitic state. This was really achieved by the removal of the silicon, although rapid cooling also played its part as the refined iron was cast and rapidly solidified into a plate. This plate was then broken up, whilst still hot, and charged into a second hearth known as the "finery." The refined iron was then treated in a very similar manner to the Lancashire hearth to effect the removal of the carbon and, to some extent, the elimination of the phosphorus. Alternatively, in some cases the refined iron was run direct into the finery whilst still in the molten condition.

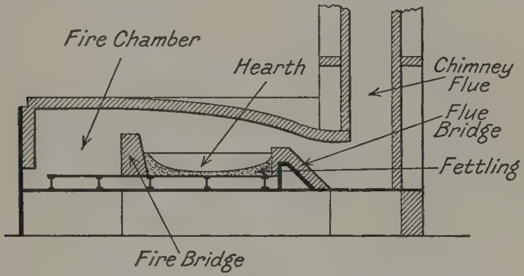


FIG. 2. SINGLE HEARTH PUDDLING FURNACE

This method of manufacture is now almost extinct, the adoption of mild steel for tin plate, etc., having been responsible for its demise.

The Puddling Process. In 1874 Henry Cort introduced the puddling process, which depended on the use of a reverberatory furnace having a melting chamber with a hollowed-out base to contain the fluid metal.

Originally, white pig was worked on a sand bottom, and purified by atmospheric oxidation. This operation was known as the "dry" puddling process—in contra-distinction to the pig boiling or "wet" method which superseded it—owing to the fact that the metal never became very fluid, and only a relatively small amount of slag was produced. Later, this furnace was worked in conjunction with a refinery and grey iron used; then a lining of iron oxide was substituted for the sand bottom and, from this, the pig boiling process was ultimately developed.

In the modern puddling process the pig iron generally used is "Forge" quality, and the oxygen necessary for the removal of its impurities is supplied in the form of various oxides of iron, which are either added during the progress of the operation or compose the lining of the hearth. These oxides are referred to as "fettling," one of the most common being the slag expressed from the puddled iron by the shingling hammer.

Grey pig and a requisite amount of hammer slag are charged into the previously heated furnace, the hearth of which has been prepared, by the skilful addition of the less fusible oxides, to form a basin. Melting down then begins. This is usually complete in about half an hour and is facilitated by the repeated turning of the pigs when the first signs of melting are apparent, and forcing the still solid portions to the surface as the operation proceeds. By the time the whole charge is in the fluid state a certain percentage of the manganese and phosphorus and a large proportion of the silicon have been oxidized, with the resultant conversion of all the carbon to the combined form. For a further period of about ten minutes a high furnace temperature is maintained and, by means of a hand-operated bar, the iron is kept constantly stirred. This brings it into intimate contact with the molten cinder which, yielding up its oxygen, completes the removal of the silicon and manganese.

The carbon is the next element to be tackled. The chimney draught is reduced, thus lowering the temperature of the furnace and filling it with a smoky flame and a reducing atmosphere. The metal thickens and, by manual exertion, is again vigorously rabbled. Thus is the carbon converted into carbon monoxide which escapes through the bath as bubbles of gas which burn, on bursting, at the surface. This state of ebullition is referred to as the " boil." During this phase the charge rises considerably in the furnace, and some of the slag is run off over the fore plate. Gradually the reaction becomes less violent, the charge again settles, and the iron takes on the familiar spongy state, having " come to nature." By possibly the hardest and hottest form of labour that man is called upon to perform, the sponge is worked up into a ball, which is then divided into five or six smaller balls of equal size and weighing about ¾ cwt. each.

These balls are taken in turn to a conveniently situated shingling hammer, and are worked down as described in connection with the Lancashire hearth. It is interesting to note that, in many cases, the steam for the shingling hammers is generated by Babcock & Wilcox boilers placed over the puddling furnaces.

On cooling off, the bars are cut into short lengths from which " piles " are built up. These may be made by simply stacking several of the short lengths on top of each other and binding them together with wire or thin strip; or by placing four lengths so as to form the four sides of a box, which is then filled with crop ends and other smaller pieces, and bound up as before. The piles are then raised to a welding heat and rolled into the desired section.

The Finished Product. No consideration of wrought iron can be complete without special mention of the slag inclusions, to which frequent reference has already been made. A large proportion of this slag is expelled under the hammer, but a certain amount remains mechanically mixed with the iron. This slag, whilst invisible to the naked eye, is most easily recognized under the microscope.

A thin section of wrought iron, cut from a plane at right angles to the direction of the rolling, is highly polished, etched with dilute acid, and examined under the microscope. The iron crystals are then revealed as irregular polyhedrons and the slag as small rounded bodies (Fig. 3). Another view of the crystals is afforded by a section, similarly prepared but cut longitudinally. Precisely the same structure is shown as regards the iron, but the slag appears as elongated streaks (Fig. 4).

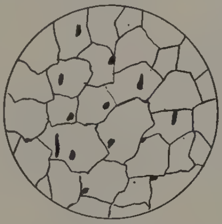


FIG. 3

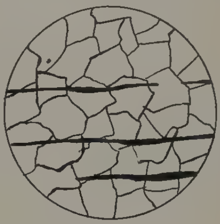


FIG. 4

From the appearance of these two sections, two inferences can be drawn—

1. That, the form of the iron crystals being the same in both, it is evident that the iron crystallized *after* leaving the rolls, otherwise the crystals would have been elongated.

2. Under the influence of the rolls the slag has been drawn out into long hair lines, parallel to the direction of rolling.

The typical fibrous appearance of a wrought iron fracture is thus accounted for by these slag lines and, since their regularity would seem to prove regular working up of the iron, the fibre can be taken as an index to quality.

Also, a little thought on the behaviour of the slag under tensile tests will furnish the reason for the fact that wrought iron is stronger " with the grain than across it."

ENGINEERING MATERIALS
TESTING AND PROPERTIES

By R. G. Batson, A.K.C., M.Inst.C.E., M.I.Mech.E.

LESSON VII

FATIGUE TESTING

As pointed out in an earlier lesson, a knowledge of the resistance of a material to repeated stresses is of great importance. Although Fairbairn made a few tests on this subject in 1864, the first extensive series of experiments was carried out by Wöhler in 1871. The conclusions reached by Wöhler were briefly—

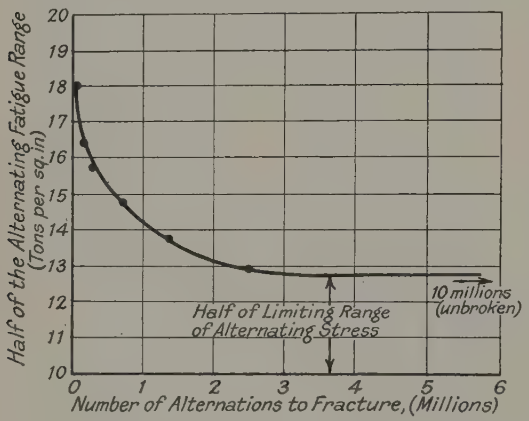


Fig. 33. Fatigue Range—Endurance Curve

1. Wrought iron may be fractured by a stress much below the ultimate stress if that stress is repeated many times.
2. For any given mean stress the number of repetitions before failure increases as the range of stress is reduced.
3. There is a limiting range of stress at which the number of repetitions is infinite.
4. As the maximum stress is increased the limiting range of stress is diminished.

If x = the maximum stress
and y = the minimum stress
then $x - y$ = the range of stress

$$\frac{x + y}{2} = \text{the mean stress}$$

A compressive stress is assumed to be negative and a tensile stress is taken to be positive, so that if a material works between a tensile stress of 15 tons per sq. in. and a compressive stress of 10 tons per sq. in., then the range of stress $= 15 - (-10) = 15 + 10 = 25$ tons per sq. in. and the mean stress

$$= \frac{15 + (-10)}{2} = \frac{15 - 10}{2} = 2\tfrac{1}{2} \text{ tons per sq. in.}$$

A material may have a limiting range of stress of, say, 80 tons per sq. in. with zero limiting mean stress, but the limiting range of stress may be only 50 tons per sq. in. with a limiting mean stress of 25 tons per sq. in.

Wöhler's conclusions have been confirmed by subsequent experimenters. Some recent research workers, by carrying out tests with hundreds of millions of repetitions, claim that with some materials there is no limiting range of stress, but that these materials will break with whatever range of stress is imposed. The number of repetitions, in extreme cases, would have to be very large.

The usual laboratory method of obtaining the limiting fatigue range is as follows—

Six test pieces are prepared from the material under test. These test pieces are highly polished so as to be free from all surface scratches. A Brinell hardness test is then made on the enlarged end of one of the test pieces by the small ball or diamond method. From the Brinell hardness number which is obtained an approximate estimate may be made of the alternating fatigue range. The ultimate strength in tons per square inch is approximately 0·22 × hardness number.

It has been found that the alternating fatigue range varies from 1·10 of the ultimate tensile strength for mild steel to 0·80 for high carbon and tool steels.

The first test piece is run in the machine with a range substantially in excess of the range estimated as above. If the estimate has been approximately correct the test piece should break after a comparatively few alternations of stress. The next piece is tested with a lower range, the amount of reduction of stress depending upon the result of the previous test. The process is repeated until a test is obtained running up to 10,000,000 alternations for steels (or 20,000,000 for non-ferrous metals and alloys) without fracture. Subsequent tests are then made with intermediate ranges of stress, the number of such tests depending upon the uniformity of the material and the accuracy required in the final result. A curve can then be drawn in the manner indicated in Fig. 33, and the limiting range of stress estimated from it with considerable accuracy. Some experimenters prefer to plot their observations logarithmically.

Fatigue tests are carried out—

1. Under reversed bending stresses,
2. Under alternate or repeated direct tension and compression.

Alternate Bending Machines. A machine for carrying out tests by this method, manufactured by W. & T. Avery, Ltd., is shown in Fig. 34. The test piece *T*, is driven by a small electric motor *M*, and has a ball bearing fixed on to its end. This ball bearing is in a special housing which is connected by a rod *D*, to a lever *L*, and counterpoise *P*. The test piece thus forms a rotating cantilever loaded at its end. When the test piece breaks, the lever falls and operates a cut-out *O* which stops the motor. The number of alternations is given by a revolution counter *C*.

The stress in the test piece is calculated from the load (= *W*), and the distance of the point of application of the load from the bottom of the radius on the test piece (= *l*). This is the maximum stress in the material due to that particular load.

If d = diameter of the test piece in inches and f = stress in the material.

$$\text{Bending moment} = Wl = \frac{\pi d^3}{32} \times f$$

$$\text{or} \qquad f = \frac{32Wl}{\pi d^3}$$

The range of stress is from $+f$ to $-f$,

i.e. $= 2f$.

The machine described above has an additional system of levers *N*, terminating in a pointer *R*. This is in connection with the "strain" method of fatigue range determination.

A strain method of determining limiting ranges of stress on Wöhler machines has been developed at the National Physical Laboratory. In this method the deflection of the end of the rotating Wöhler test piece is measured under increasing loads by the movement of a spot of light reflected from a mirror fixed on the end of the test piece. If the amount of this deflection is plotted against the load, a curve is obtained which is very much like the stress-strain curve in the tensile test. There is a definite departure from proportionality. The stress corresponding to the load at this point of departure is found to be equal to half of the alternating limiting range of stress. The method as at present carried out is of doubtful value except for the more ductile materials.

The pointer in the Avery machine takes the place of the optical system used at the National Physical Laboratory.

Alfred J. Amsler makes a fatigue machine on the Wöhler principle. A somewhat similar machine, except that the test piece is a rotating bar (7·5 mm. diameter and 75 mm. long) under uniform bending moment, is supplied by C. F. R. Giesler, Ltd., Holborn, London.

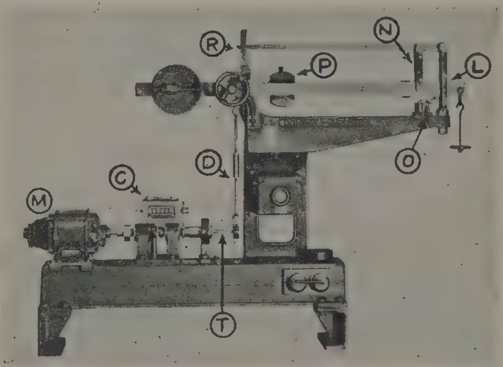


FIG. 34. WÖHLER FATIGUE TESTING MACHINE
(By W. & T. Avery, Ltd.

Alternate Direct Tension and Compression Machines. Dr. J. H. Smith designed a machine of this type in which an alternating range of stress is applied to the test piece by means of an unbalanced fly-wheel. This gives a range of stress with zero mean stress and, in order to obtain unequal tensile and compressive stresses, a steady stress is superimposed on the alternating stress by means of a calibrated spring.

Several makes of machine apply the load by means of the pull of electro-magnets excited by an alternating current. The best known of these machines is that designed by Dr. B. P. Haigh, and is made by Bruntons, Musselburgh, Scotland. The upper end of the test piece is connected to the frame of the machine, while the lower end is attached, through suitable framework, to the armature of a two-phase electro-magnet. The electro-magnet is excited by an alternating current and the whole is arranged so that the test piece is subjected to alternate tension and compression. Suitable provision is made to eliminate all possible sources of error, such as the inertia of the moving parts, etc. A superimposed steady load can be applied by means of a spring in order to vary the value of the mean stress. The machine works at 2,000 stress cycles per minute.

C. F. R. Giesler, Ltd., supply an alternating direct tension and compression machine working at 30,000 alternations per minute, and operated by applying the resonance action of an oscillating system together with the use of electro magnetic exciter forces.

TABLE III

CARBON STEELS (NORMALIZED)

Material	Yield Stress Tons per sq. in.	Ultimate Stress Tons per sq. in.	Elongation %[2]	Reduction of Area %
0·02% Carbon[1]	11·8	22·1	45	69
0·13% "	13·5	22·4	49	71
0·17% "	16·8	28·5	43·5	68
0·24% "	17·6	32·1	40	60
0·26% "	24·5	35·5	32	59
0·45% "	27·0	44·0	27	55
0·65% "	31·2	53·2	22·5	44

[1] Armco Iron.

[2] The elongation was measured on a gauge length of four times the square root of the cross-sectional area of the test piece.

TABLE IV

ALLOY STEELS

N = Normalized.
T = Tempered at.
OQ = Oil Quenched.
AH = Air Hardened.
WQ = Water Quenched.

Material Per cent Composition	Heat Treatment	Yield Stress	Ultimate Stress	Elongation[1] %	Reduction of Area %	Izod. ft.-lb.	Brinell No.
		Tons per sq. in.					
C. 0·15, Ni. 2·60 Nickel steel	N–880° C.	24·0	32·0	37	69	95	143
	OQ–880° C.	—	38·0	30	70	100	179
	WQ–880° C.	—	50·0	23	60	51	241
	OQ–880°C. T–600° C.	26·0	35·0	35	74	107	163
C. 0·37, Ni. 3·65 Nickel steel	N–860° C.	27·0	44·0	28	56	36	196
	OQ–860° C.	—	116·0	10	—	6	495
	WQ –860° C.	—	124·0	9	25	4	514
	OQ–860° C. T–600° C.	44·0	52·0	25	65	73	241
C. 0·31, Ni. 3·27, Cr. 0·82 Nickel chromium steel	N–820° C.	—	76·0	15	40	6	341
	OQ–820° C.	—	116·0	14	37	11	495
	OQ–820° C. T–600° C.	54·0	60·0	22	62	59	285
C. 0·32, Ni. 3·70, Cr. 1·42 Nickel chromium air hardening steel	AH–820° C.	—	114·0	11	39	17	477
	OQ–820° C.	87	91·0	13	52	17	418
	OQ–820° C. T–650° C.	47	57	24	67	86	262
C. 0·54, Si. 1·95, Mn. 0·94 Silico manganese steel	WQ–870° C. T–500° C.	85	91	8	22	14	415
C. 0·55, Cr. 1·16, Vd. 0·27, Ni. 0·10 Chrome vanadium steel	OQ–850° C. T–550° C.	82	85	7	32	20	415

[1] The elongation was measured on a gauge length of four times the square root of the cross-sectional area of the test piece.

ECHANICAL PROPERTIES OF MATERIALS
(Values)

The mechanical properties of steel depends to large extent upon the previous mechanical or eat treatment to which the steel has been subected. Cold working usually leaves the steel ith internal stresses which are released by ormalizing. For experimental work it is ssential that the steel should be correctly ormalized or otherwise heat treated, and this eatment should always be stated in any ublished reports of research work.

The ultimate tensile strength of ordinary traight carbon steel increases as the carbon ontent increases. This increase is accompanied y a reduction of the elongation, and of the eduction of area at fracture. Typical results n normalized carbon steels are given in able III.

The tensile properties of alloy steels can be ubjected to considerable variation and improveient by suitable heat treatment. The properes of a few typical alloy steels are given in able IV. Some results on non-ferrous alloys re given in Table V for comparison.

The modulus of direct elasticity or Young's odulus *E* is nearly the same in tension as in ompression, and it is practically independent f the carbon content. The values range etween $28{\cdot}5 \times 10^6$ lb. per sq. in. and $31{\cdot}0 \times$ 0^6 lb. per sq. in., but the majority of steels ary between much closer limits, viz., $29{\cdot}5 \times 10^6$ and $30{\cdot}0 \times 10^6$ lb. per sq. in. A few typical values of *E* together with the corresponding values of the modulus of rigidity *N* are given in Table VI.

TABLE V

NON-FERROUS METALS AND ALLOYS

Material	Yield Stress Tons per sq. in.	Ultimate Stress Tons per sq. in.	Elongation %[1]	Reduction of Area %
80/20 rolled nickel-chromium	36·7	52·1	37	56
70/30 rolled nickel-copper	25·0	37·4	46	70
Rolled nickel	9·4	33·8	44	57
Rolled copper	—	15·2	52	57
Rolled monel metal	22·3	44·6	30	50
Duralumin bar	16	25	15	20

[1] The elongation was measured on a gauge length of four times the square root of the cross-sectional area of the test piece.

TABLE VI

Material	Modulus of Direct Elasticity Lb. per sq. in.	Modulus of Rigidity Lb. per sq. in.
Armco iron	$29{\cdot}8 \times 10^6$	$11{\cdot}6 \times 10^6$
Rolled nickel	$29{\cdot}5 \times 10^6$	$11{\cdot}2 \times 10^6$
80/20 rolled nickel chromium	$31{\cdot}6 \times 10^6$	$12{\cdot}6 \times 10^6$
70/30 rolled nickel-copper	$26{\cdot}0 \times 10^6$	$10{\cdot}2 \times 10^6$
Monel metal	$26{\cdot}0 \times 10^6$	$10{\cdot}0 \times 10^6$
Rolled aluminium	$10{\cdot}0 \times 10^6$	$3{\cdot}8 \times 10^6$
Steels	$29{\cdot}5 \times 10^6$ to $30{\cdot}0 \times 10^6$	$11{\cdot}5 \times 10^6$ to $12{\cdot}0 \times 10^6$
Rolled copper	$15{\cdot}0 \times 10^6$	$5{\cdot}6 \times 10^6$

(CONCLUSION)

STRUCTURAL STEELWORK

By W. Arnold Green, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON I

STRUCTURAL STEEL

Rolled Sections. There are available for the designer of structural steelwork many types of sections which can be obtained in many sizes, and usually in lengths up to 40 ft. and in greater lengths at extra cost.

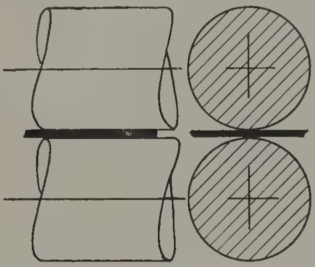


Fig. 1

In the finishing process of rolling, *plates* are produced by passing the hot metal (which has been reduced to a suitable size from a white-hot ingot by squeezing it between revolving grooved rolls in a *blooming* or *cogging* mill) between cylindrical rolls, which are brought closer together until the required thickness is obtained (see Fig. 1)[1]. Plates produced in a *universal* mill have the edges also rolled square and parallel. For *rounds* and *squares*, grooves of the required shape are cut in the finishing rolls as indicated in Fig. 2. Fig. 3 illustrates the final process in the rolling of *flats* and *angles*, and Fig. 4 that of *joists*, where the grooves forming the section are cut alternately in the upper and lower rolls. Other flanged sections such as *tees* and *channels*, are similarly produced.

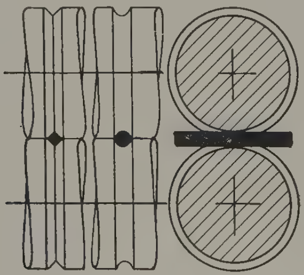


Fig. 2

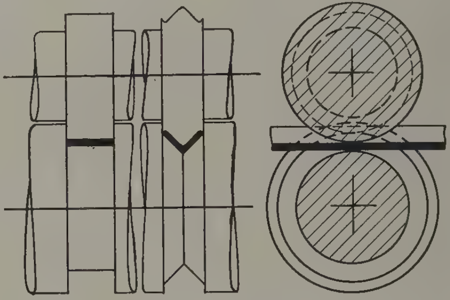


Fig. 3

In the " Grey " mills for rolling *broad flange beams*, three pairs of rolls as shown in Fig. 5 are used in the finishing process. The flanges are made without the taper necessary for joists manufactured with ordinary rolls.

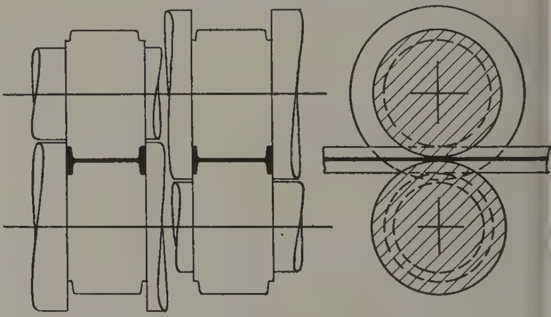


Fig. 4

Angle sections are also rolled with the edge of the long side thickened; these *bulb angles* are mostly used in shipbuilding.

Another valuable section much used in bridge work is *steel troughing*, which is riveted as shown in Fig. 6. The total depth may be increased by riveting a plate between the two sections as indicated in the left of the figure.

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[1] The drawings for Figs. 1-5 are reproduced by the kind permission of Messrs. R. A. Skelton & Co., from their handbook, No. 20. They are merely diagrammatic and are drawn to different scales.

Standard Sections. Excellent handbooks are issued by various makers giving full particulars of the sections available. Tables of properties of *British Standard Sections* are published by the *British Engineering Standards Association* (hereafter referred to as B.E.S.A.).

Quality of Steel. The method of manufacture and chemical composition need not concern the structural engineer if what he is going to use satisfies the requirements of British Standard Specification, No. 15 (1912). He will be assured of this by maker's certificate and test sheets, or he may have independent tests made by a testing engineer.

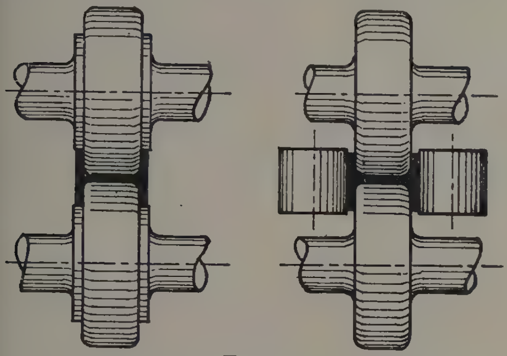

FIG. 5

Two grades of steel are dealt with in the specification, of which the latter is not intended for bridges, plates $\frac{1}{4}$ in. thick and over and rivet bars. The breaking tensile stress measured on the initial section must be from 28 to 33 tons per sq. in. with a minimum elongation of 20 per cent on a gauge length of 8 in. (for rivet bars, 25 to 30 tons per sq. in. with a minimum elongation of 25 per cent).

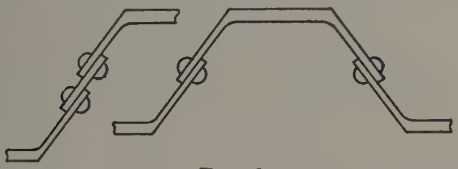

FIG. 6

For further tests, allowable *length* and *weight margins*, etc., the reader is referred to B.E.S.A., No. 153, in which the specification is published in Appendix I.

Heavy Sections. By lifting the rolls extra material is added as indicated by the white portions of the sections shown in Fig. 7. Makers, however, require a sufficiently large order for one particular section to justify such a change.

Special Steels. Similarly it is sometimes possible to have sections rolled with a special steel. Thus, in *Engineering News Record* for 16th Sept., 1926, a considerable saving is reported in a large roof by the use of *silicon steel* and higher working stresses than for standard steel.

Girder Design. When the loads to be carried by a girder are known, the bending moments and shears due to the loads can be found on the principles set out in "Applied Mechanics."

If the maximum bending moment is divided by the working stress, a value is obtained for the required *section modulus* of the girder, i.e. the quotient of its *moment of inertia* divided by the distance of the neutral axis from the stressed edge. Care must be taken in the choice of units. Thus, if the working stress is expressed in tons per square inch, and the section modulus in $(\text{inches})^3$, the bending moment must be expressed in inch-tons.

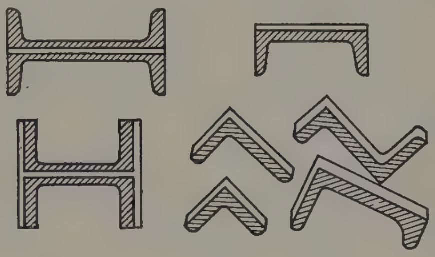

FIG. 7

Weight of Girder. A higher value of the section modulus will be required, as the weight of the girder itself produces bending moment.

As, however, the loads carried can often only be rough approximations, great accuracy in estimating the weight of the girder itself is not justified in the preliminary stages, except for large spans, where the dead load of the girder may be a considerable part of the total load. For commercial reasons and erection purposes it is necessary to have a close estimate of the total weight of steel when the design is completed. If this differs largely from the assumed weight, adjustment may be necessary.

Structural steel weighs 490 lb. per cub. ft., or 40·8 lb. per sq. ft. 1 in. thick, which is equivalent to 3·4 lb. per foot run for each square inch of cross-section. If the beam is encased in concrete the weight may be taken as 1 lb. per sq. in. of cross-section, i.e. it may be assumed to weigh 144 lb. per cub. ft.

Simple Sections. If possible, for the sake of economy, a section with the required section modulus should be selected from rolled beams (which can be obtained up to 24 in. deep if of British standard section, and up to about 40 in. deep if "Broad Flange Beams") to obviate the large amount of work which plating (i.e. the attachment of flat plates to the flanges by means of rivets) entails in the shops.

It is sometimes advisable to use a pair of unplated joists, connected by two bolts (or one in the case of shallow sections) through the webs and through *cast-iron separators*, which are shaped to bear on the flanges top and bottom to make the two joists act as one. The separators are placed at the ends under concentrated loads, and at centres about five times the joist depth.

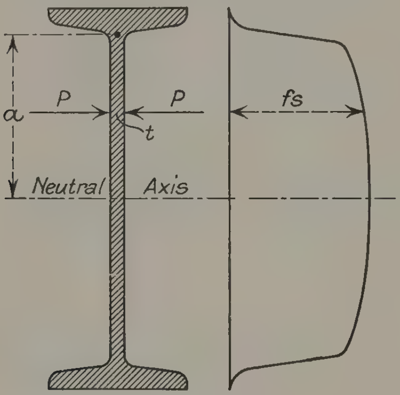

FIG. 8

Short Spans. For short spans carrying heavy loads, the deciding factor in the selection of a suitable section may be not the bending moment, but the shear. In the safe loads tabulated in steel makers' handbooks, it will be noted that often no loads are given for short spans because the uniformly distributed loads producing the working stress at the edge of the beam (say, 7½ or 8 tons per sq. in.) produce a stress in the web higher than that allowable (say, 4 or 5 tons per sq. in.); the latter figure is that given in Part III of B.E.S.A., No. 153, the whole of which publication the student is recommended to study.

Shear Stresses. The shear stress is usually calculated by dividing the total shear by the total area of the web, i.e. the product of the full depth of the joist or channel and the web thickness.

The shear stress is not constant throughout the depth, but for a rolled steel joist varies, a
indicated in Fig. 8, the stress at any horizonta
web section *PP* equalling $S \times A \times a \div t \times$
where S = total shear on vertical section, A =
area of the shaded portion between *PP* and th
edge of the section, a = distance of its centr
of gravity from the neutral axis, t = thicknes
of web at *PP* and I = the moment of inerti
of the whole section.[1]

Diagonal Stresses. Consider a small squar
element of the web $ABCD$ of thickness t sub
jected to a shear stress f_s along BC and D
(see Fig. 9). If it is in equilibrium there mus
be a similar stress along BA and DC to preven
rotation. These horizontal shear stresses caus
the changes in the horizontal compressive an
tensile stresses resisting the bending moment
Pure shear stresses along the edges of the squar
$ABCD$, such as those occurring at the neutra
axis, are equivalent to a compressive stress o
the same intensity parallel to the diagonal BD
and a tensile stress of the same intensit
parallel to the diagonal AC. When shea
stresses are combined with direct tensile o
compressive stresses normal to the sides of th
square, the resulting *principal stresses* are modi
fied in intensity and direction; but in the we
the stresses are always equivalent to an incline
compressive stress with a tensile stress at righ
angles to it, the direction at the neutral axi
being, of course, at 45° to that axis.

Web Buckling and Stiffness. The diagona
compressive stress in the web tends to caus
buckling, though the diagonal tension has
restraining effect on this buckling tendency.

In rolled steel joists the web rarely require
strengthening for diagonal compression, bu
stiffeners of angle section are sometime
riveted to the webs under heavy concentrate
loads and at bearings. A discussion of we
stiffeners for built-up sections must be left t
the next lesson.

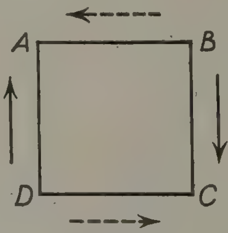

FIG. 9

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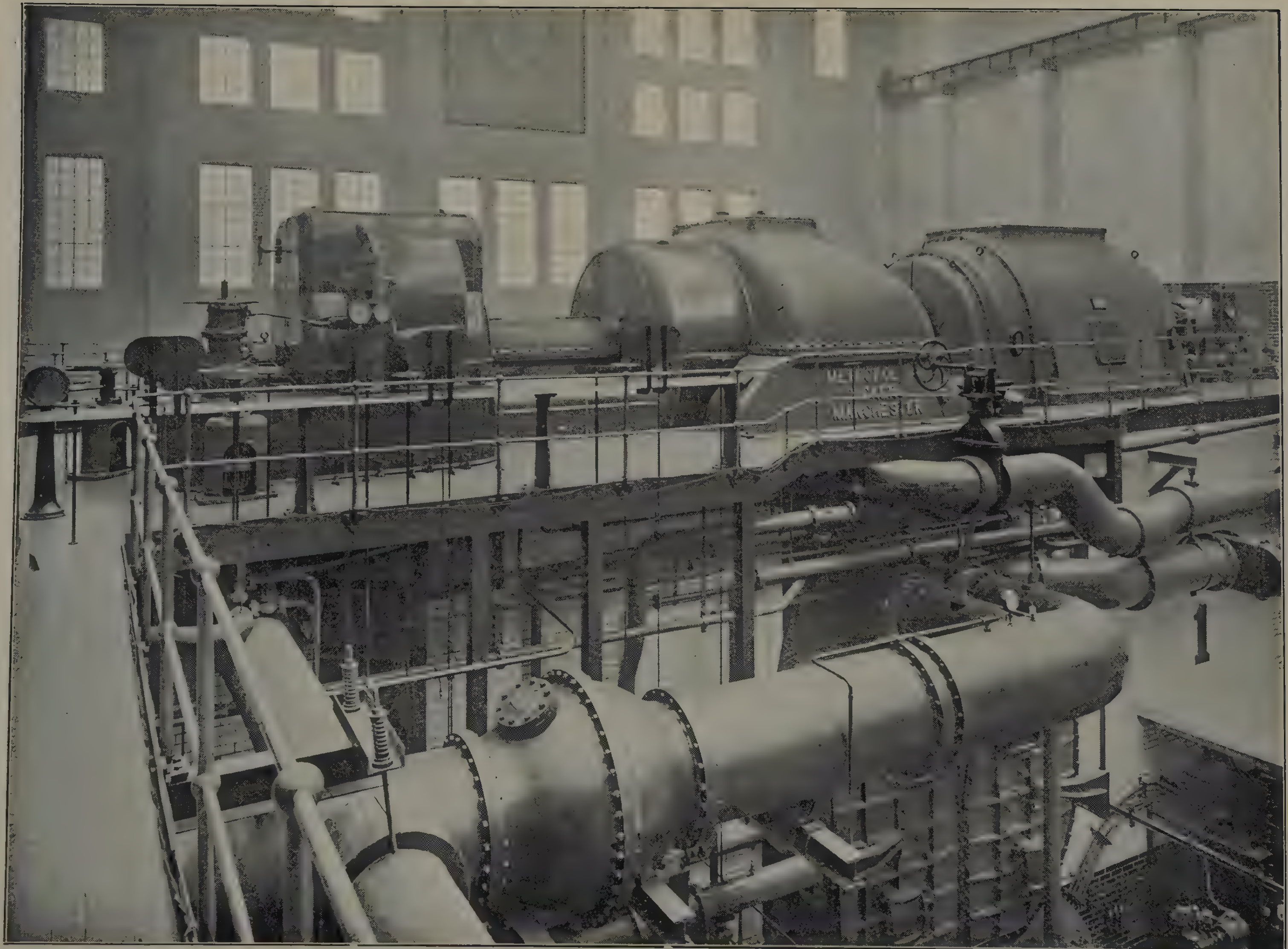
[1] See also Lessons X and XI of "Applied Mechanics.



45,000 B.H.P., 1,500 R.P.M. TWO-CYLINDER STEAM TURBINE

STEAM TURBINES

By W. J. Kearton, M.Eng., A.M.I.Mech.E., A.M.I.N.A.

LESSON VI

IMPULSE TURBINE CONSTRUCTION

Impulse turbines, as constructed to-day may be classified roughly in the following manner according to size, type, and mechanical arrangement.

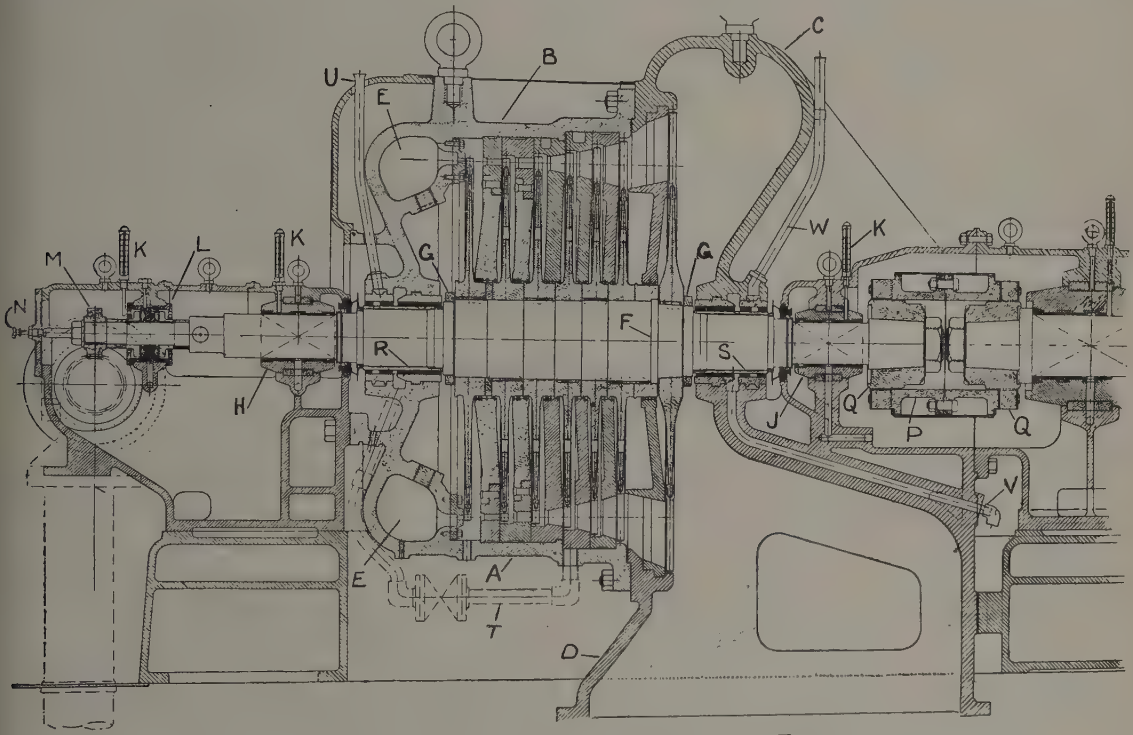


Fig. 22. Seven-stage High Pressure Impulse Turbine
6,000 kw. at 3,000 r.p.m.
(Messrs. Fraser & Chalmer's Engineering Works)

(*a*) Simple impulse turbines of the de Laval type, coupled through reduction gearing to fans, pumps, or dynamos. Generally used for auxiliary purposes.

(*b*) Single-stage turbines of the velocity-compounded type having two or three rows of moving blades, and directly coupled to fans, pumps, or dynamos. These are usually small machines used for auxiliary purposes.

(*c*) Multi-stage turbines of powers varying from 200 to 1,500 kw. having either single-row wheels, two-row wheels, or a two-row wheel in the first stage followed by single-row wheels in the remaining stages, coupled by reduction gearing to pumps or generators.

(*d*) Multi-stage turbines of powers varying from 1,000 to 10,000 kw. or more, having usually a two-row wheel in the first stage, and single-row wheels in the remaining stages, but sometimes single-row wheels throughout, and coupled direct to A.C. generators. Such turbines operate at fairly high speeds, 2,400, 3,000, 3,600 r.p.m., depending on the frequency of the alternating current.

For if f = frequency, i.e. number of cycles per second.
N = revolutions per minute.
n = number of poles.

Then in one revolution of the rotor, the number of cycles $= \frac{n}{2}$.

$$\therefore \text{ Frequency } f = \frac{N}{60} \times \frac{n}{2}.$$

$$\therefore N = \frac{60f}{\frac{n}{2}} \text{ rev. per minute.}$$

Thus in a two-pole machine with a frequency of 50 cycles per second, $N = 3{,}000$.

(*e*) Turbines of powers varying from 15,000 to 50,000 kw. of type (*d*), but modified at the low pressure end in order to cope with the very large volume of exhaust steam.

(*f*) Impulse turbines of moderate power and of special construction, such as *mixed-pressure* and *heat-extraction turbines*.

(*g*) Marine turbines. Similar to (*d*), but almost invariably geared down through single- or double-reduction gearing to the propeller shaft.

The following descriptions will relate to types (*d*) and (*e*).

A sectional elevation of a seven-stage turbine fitted with single-row wheels throughout is shown in Fig. 22. The machine was constructed by Messrs. Fraser & Chalmer's Engineering Works, and is designed to develop 6,000 kw. at a speed of 3,000 r.p.m. when steam is supplied at 200 lb. per sq. in. gauge, superheated 150° F., and when a vacuum of 28·25 in. (30 in. barometer) is maintained in the condenser.

The turbine casing is made in four parts, the two halves *A* and *B* being of cast steel, and the two halves *C* and *D* at the low pressure end being of cast iron. It will be observed that the steam chest *E* into which the steam flows after passing through the governor valves is an integral part of the casing. The temperature of dry and saturated steam at 200 lb. per sq. in. is 388° F. and, since the superheat is 150° F., the total temperature of the steam is about 540° F. When cast iron is subjected to high temperatures it suffers a kind of deterioration, known as "growth," and practical experience with steam turbines seems to show that 450° F. is about the safe limit of temperature for cast iron. Hence the reason for adopting cast steel which is not only very much stronger, but does not suffer this particular type of deterioration. By the expansion of the steam which occurs in the first stage nozzles, particularly when a two-row wheel is used in the firs stage, the temperature is so much reduced tha cast iron may, as a rule, be used with safety Many turbine builders, therefore, by isolatin the live steam in a special cast steel steam chest, are thereby enabled to construct the whol casing of cast iron.

To return to our description, steam enters the turbine from the chest *E* from whence i flows through nozzle plates bolted to a machined face on the chest. After passing through the first stage blades it flows direct into nozzles placed in the first diaphragm, in which it undergoes further expansion. As the pressure is diminished the total volume of steam increases and the necessary increase of area in the low pressure nozzles is obtained by making the length of arc covered by the nozzle openings increase until it covers the whole circumference, by increasing the radial height of the nozzles and finally by increasing the nozzle angle. In consequence, the blades are made much longer at the low pressure end than at the high pressure end.

The rotor discs are a force fit on the stepped shaft, and are located by a single collar *F* on the spindle. The discs are secured by the nuts *G*. The shaft is supported by journal bearings *H* and *J*, and its axial position is fixed by the adjusting block *L* which is of the Michell pivoted pad type. The gear wheel *M* drives the governor and oil pump. The poker gauge *N* serves to indicate the approximate axial adjustment of the rotor.

The turbine and its generator are coupled together by a flexible coupling which is designed to make allowances for any slight lack of alignment between the two shafts. The complete coupling comprises two claw pieces *Q* keyed to the turbine and generator shafts, and a sleeve *P*.

Leakage of steam out of, and of air into, the turbine casing is minimized by the labyrinth glands *R* and *S*. These will be described in detail afterwards. It may be pointed out, however, that the "pocket" of the gland *R* is coupled by the pipe *T* to a cell of the turbine, in which the pressure of the steam is normally just a little greater than atmospheric. Thus, the high pressure steam which leaks past the first section of the gland may do work by further expansion in the last three stages of the turbine. In order to prevent air from leaking into the low pressure end of the turbine the gland *S* is "sealed" by steam which is supplied via the pipe *V*. The waste steam from the glands is allowed to escape into the engine-room through the vapour pipes *U* and *W*. Some form of baffle or deflector is usually attached to the shaft close to the gland mouth, so as to prevent steam from blowing on to the bearing or its housing.

A longitudinal section through a typical Metropolitan-Vickers high pressure turbine is shown in Fig. 23. In this example, there is a two-row wheel in the first stage followed by six single-row wheels. In other respects, the turbine is constructed on the same general lines as the machine described above with the following variations of detail.

The steam chest is a separate casting bolted to a facing machined on the outside of the turbine casing. The nozzle blocks *c* communicate by means of a distance piece with the chest so that no live steam comes into contact with the turbine casing.

The high pressure gland is a combined labyrinth and water seal, while the low pressure gland is a simple water seal.

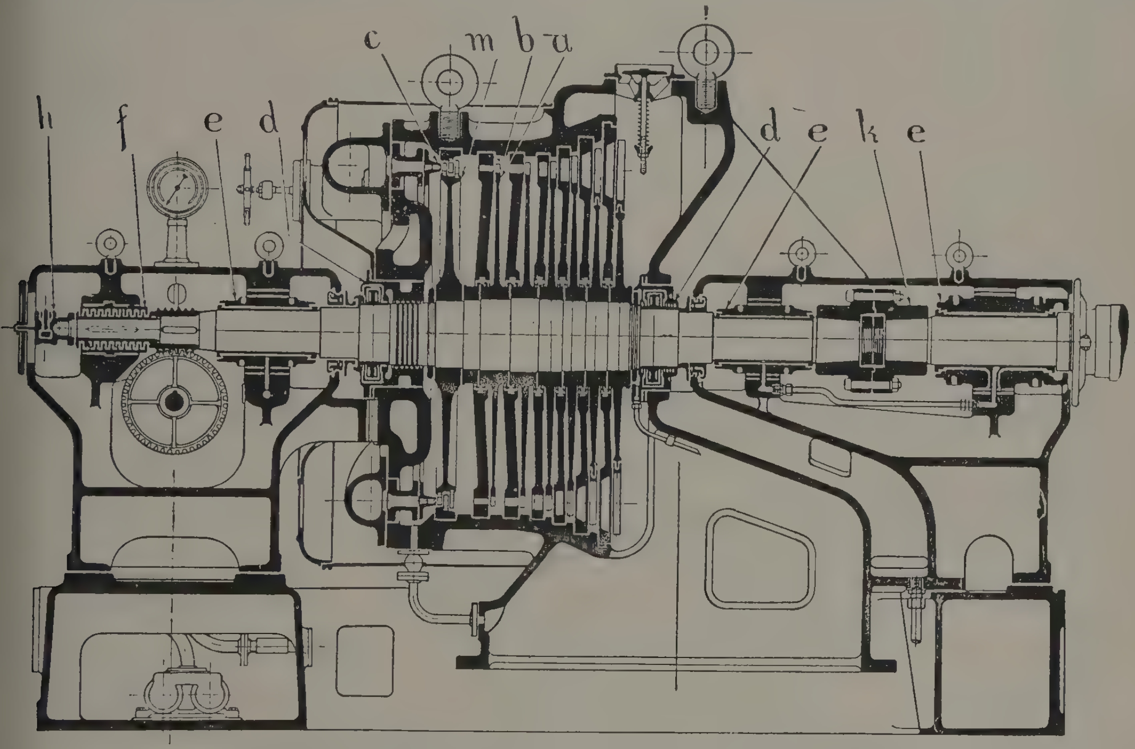


Fig. 23. Seven-stage High Pressure Impulse Turbine
(Messrs. The Metropolitan-Vickers Electrical Co., Ltd.)

The combination of a two-row velocity-compounded wheel with single-row wheels is further exemplified in Fig. 24, which shows a section through a high pressure turbine constructed by the British Thomson-Houston Co., of Rugby. The special feature of the turbines built by this firm is the system of governing adopted. Reference will be made to this later, but it may be stated briefly that instead of having one or two throttle valves controlling the supply of steam to individual groups of nozzles, there are, in these machines, from five to eight such valves, each controlling the supply of steam to one or more nozzles. The valves *N* are operated in turn by means of the rotating cams *M*, according to the load on the turbine. The camshaft is rotated by an oil-pressure operated servo motor controlled by the speed governor. The adoption of this "nozzle-control" system of governing results in a reduced steam consumption at light loads.

The axial position of the rotor is determined by the multi-collar adjusting block *H*. This is in halves, and the end remote from the turbine is screwed externally to fit a large nut attached to the journal bearing *B*. A collar on the block *H* is machined as a worm wheel, and the whole block may be rotated by means of the worm *J*. In this manner, the axial position of the rotor may be nicely adjusted without the necessity of dismantling the bearing and fitting liners, as is usual.

Two-cylinder Turbines. The illustration facing page 825 shows a two-cylinder turbine of 45,000 b.h.p. maximum continuous rating capacity, as installed at the Lister Drive Power Station, Liverpool, by The Metropolitan-Vickers Electrical Co., Ltd. This machine has been designed to run at a speed of 1,500 r.p.m. The development of this type of unit has been necessary for two reasons—

EXPLANATORY NOTES

A = Shaft.
B = Turbine end bearing.
C = Claw coupling.
D = Oil guards.
E = Shaft packings.
F = Diaphragm packings.
G = Screw threads for attaching hydraulic jack.
H = Adjusting block.
J = Operating gear for adjusting block.
K = Emergency governor.
L = Worm drive for governor and oil pump.
M = Cams for operating nozzle valves.
N = Nozzle control valves.
O = First-stage nozzles.
P = First-stage wheel.
Q = Intermediate guide blades.
R = Diaphragms.
S = Bucket wheels.
T = Tachometer.
U = Steam end pedestal.
V = Vapour pipes from glands.
W = Turbine centre bearing.
X = Centre pedestal.
Y = Generator centre bearing.

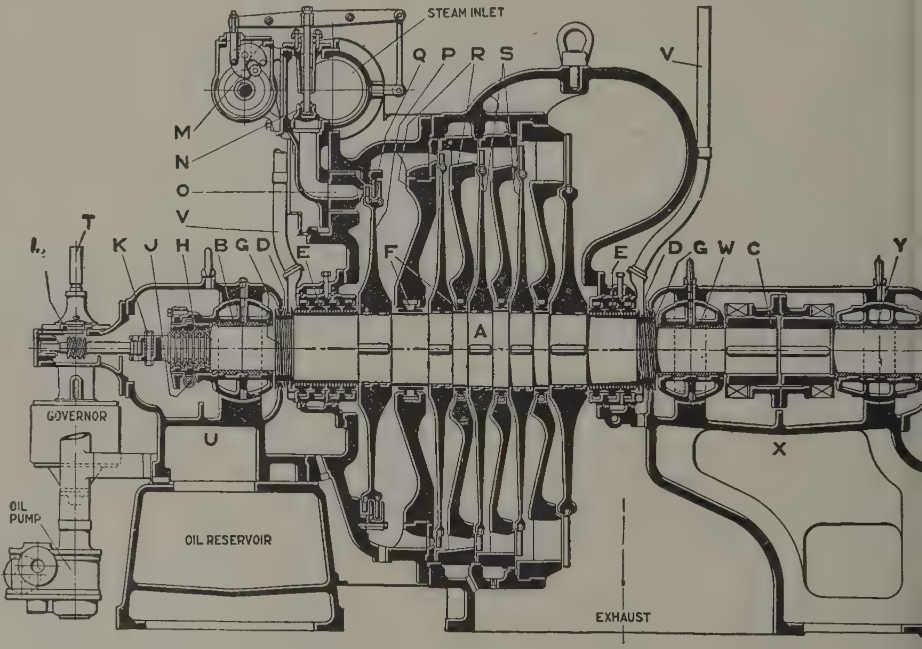


FIG. 24. FIVE-STAGE HIGH PRESSURE IMPULSE TURBINE
(Messrs. The British-Thomson-Houston Co., Ltd.)

1. To meet the increasing demand for higher steam pressures and temperatures.

2. To give a higher total efficiency than is possible with a single-cylinder turbine.

As illustrated, the unit comprises a separate high pressure and low pressure cylinder. This arrangement offers many advantages over th single-cylinder type unit which may be summarized as follows. The disc of the rotatin element in the high pressure unit can be o relatively small diameter with the result tha the steam velocities can be reduced, thu reducing the frictional losses of the steam on th blading.

The peripheral speed of the discs is low whic means lighter construction, and the stresses se up in them will, therefore, be of small order giving a factor of safety correspondingly large.

The high pressure unit can be constructe throughout of steel, thus confining the steam i its highly superheated state, leaving the lo pressure cylinder subjected to steam of relativel low pressure and temperature.

Regenerative feed-water heating to 250° o 300° F., by taking steam from various stage of the turbine to three or four feed-wate heaters, is now adopted still further to increas the fuel economy.

APPLIED MECHANICS

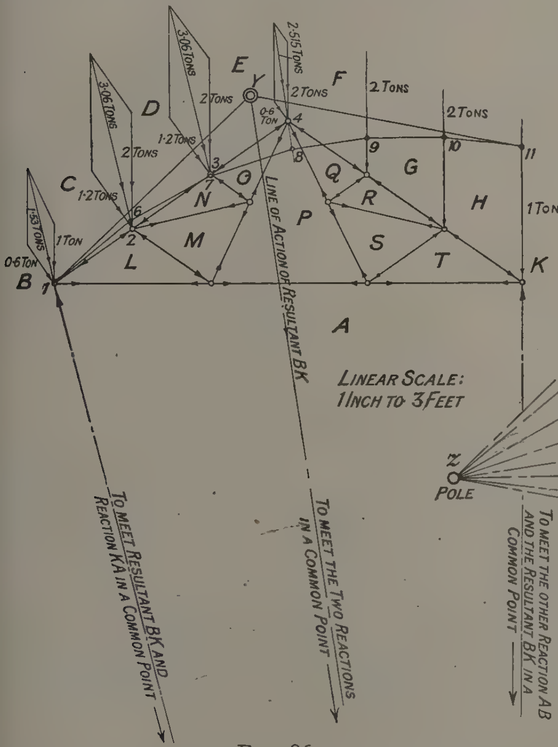
By Geo. W. Bird, Wh.Ex., B.Sc., A.M.I.Mech.E., A.M.I.E.E.

LESSON XVII

GRAPHIC STATICS—(*contd.*)

Roof Truss with Wind Load

Since every roof is, at times, acted upon by the wind, in addition to the constant vertical loads, we must be prepared to deal with this case. Let us consider the roof truss shown in Fig. 86; the span of the roof being 36 ft. with a pitch angle of 35°. The roof carries a number of dead vertical loads, and, in addition, is acted upon by the wind on the left-hand side, the normal components of the wind being 0·6, 1·2, 1·2 and 0·6 tons acting at the four joints respectively, as shown. Let us assume that the right-hand support is so constructed as to permit of horizontal movement only, which, being assumed frictionless, gives a vertical reaction at this end. The roof, however, requires anchorage and this will be provided at the left-hand support; the magnitude and direction of this reaction is to be determined. Having set out the roof truss to a suitable linear scale of, say, 1 in. to 3 ft., as shown in Fig. 86, we determine the resultant applied load at each of the four joints, 1, 2, 3, and 4, by means of the parallelogram of forces. We have now reduced the applied loads to the following: 1·53, 3·06, 3·06, 2·515, 2, 2 and 1 tons respectively and our next step is to determine the magnitude and line of action of the single resultant force, which may replace these applied loads. Selecting a suitable load scale of, say, 1 in. to 1 ton, we can set down the line of loads, *bcdefghk*. From any point *z*, taken as pole point, draw *Zb*, *Zc*, *Zd*, *Ze*, *Zf*, *Zg*, *Zh*, and *Zk*. We can now obtain the funicular polygon by drawing,



0·6 Ton
1·53 Tons
1 Ton
B
1
C
1·2 Tons
3·06 Tons
2 Tons
D
2
6
L
M
N
1·2 Tons
3·06 Tons
2 Tons
3
7
O
E
Y
2·515 Tons
0·6 Ton
2 Tons
4
8
F
2 Tons
9
Q
P
R
G
2 Tons
10
11
H
1 Ton
S
T
K
A
Line of Action of Resultant BK
Linear Scale: 1 Inch to 3 Feet
To meet Resultant BK and Reaction KA in a common point
To meet the two reactions in a common point
To meet the other reaction AB and the resultant BK in a common point

Fig. 86

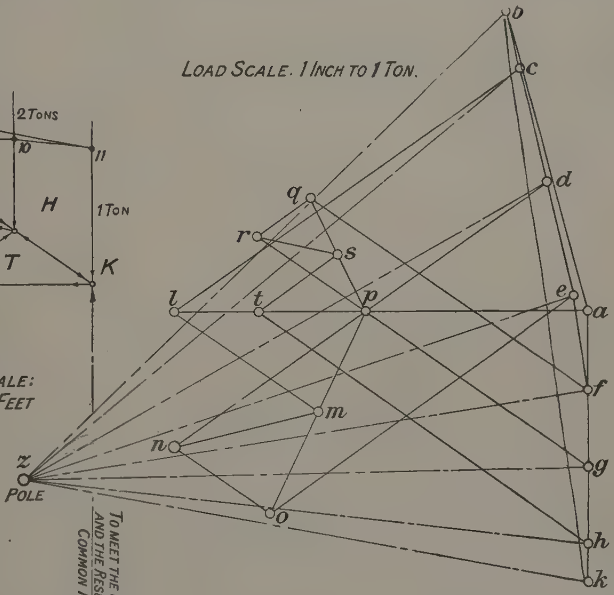


Fig. 87[1]

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[1] Note: These figures are ⅕ the size of the original

from point 1 (the left-hand point of support), the lines 1, 6 parallel to Zc; 6, 7 parallel to Zd; 7, 8 parallel to Ze; 8, 9 parallel to Zf; 9, 10 parallel to Zg; and 10, 11 parallel to Zh. The closing lines $1Y$ and $11Y$ are drawn parallel to Zb and Zk respectively. Y is a point on the line of action of the resultant of the applied loads, this resultant being the force bk, shown on the force diagram, Fig. 87. It will be seen that the roof truss is in equilibrium under the action of three forces, viz., the two reactions KA and AB, together with BK, the resultant of all the applied loads, so that our problem is now reduced to the simple case of the triangle of forces, and hence these three forces must act at a point. This condition enables us to determine the line of action of the reaction AB, since it must pass through the point of intersection of the lines of action of the other two forces, and also through the left-hand point of support of the roof truss. The point of intersection is too remote to be shown on the diagram, but its position can be calculated by simple proportion, and the angular position of the reaction AB fixed. Returning to our force diagram, Fig. 87, draw ba parallel to the reaction AB, and ka parallel to the reaction KA. We can now proceed with the force diagram by drawing al parallel to LA, and cl parallel to CL. Before any further graphical construction is made, we will calculate the force pa in the horizontal tie bar PA. The method of making this calculation was explained in Lesson XVI. Taking moments of forces about the joint 4, and considering the right-hand side of a section taken through this joint and the horizontal tie bar PA, we have:—

$$(FG \times 2 \text{ in.}) + (GH \times 4 \text{ in.}) + (HK \times 6 \text{ in.}) + (AP \times 4{\cdot}2 \text{ in.}) = (AK \times 6 \text{ in.})$$

or, $$(2 \text{ tons} \times 2 \text{ in.}) + 2 \text{ tons} \times 4 \text{ in.}) + (1 \text{ ton} \times 6 \text{ in.}) + (AP \times 4{\cdot}2 \text{ in.}) = (7{\cdot}0835 \text{ tons} \times 6 \text{ in.})$$

$$\therefore\ AP = 5{\cdot}833 \text{ tons.}$$

In Fig. 87, mark off $ap = 5{\cdot}833$ tons to scale draw pm parallel to PM and lm parallel t LM. The completion of the force diagram Fig. 87, presents no further difficulties. Th forces acting upon the various bars can b scaled off, Fig. 87, and tabulated as under—

Bar	Force in Tons	Bar	Force in Tons	Bar	Force in Ton
AL	− 10·82	GR	+ 10·56	OP	− 5·83
AP	− 5·833	HT	+ 10·56	PQ	− 3·3
AT	− 8·63	LM	+ 4·58	PS	− 1·65
CL	+ 11·08	NM	− 3·86	QR	+ 1·74
DN	+ 11·96	NO	+ 3·02	RS	− 2·15
EO	+ 9·77	MP	− 2·9	ST	+ 2·55
FQ	+ 8·86				

+ indicates a strut; − indicates a tie.

TO DRAW A FUNICULAR POLYGON THROUGH TWO GIVEN POINTS

Determination of Reactions

In the problems which we have, as yet, solve it has not been necessary to draw the *funicula polygon* to pass through a certain fixed poin or points. There are occasions, however, whe it is necessary to draw the funicular polygon t pass through either, (1) two, or (2), three fixe points, and the methods will now be considere Let us consider a system of, say, four forces AB, BC, CD, and DA, which is in equilibrium CD and DA being parallel forces. Set down, t scale, the forces ab and bc, Fig. 89. Select an pole p, and draw ap, bp, cp, and ac; ac bein the resultant, or ca the equilibrant of ab an bc. Let s and t be points on the lines of actio of DA and CD respectively. Since ca is th total equilibrant, it may be divided into tw correctly proportioned parts, one of which, da acts through the point s, and the other, cd through the point t. We require to determin these two parts, cd and da. Draw the funicula polygon, $efgh$, as shown in Fig. 88; and i Fig. 89, draw pd parallel to the closing side e of the funicular polygon. Then, to the forc scale chosen, cd represents one of the reaction CD, and da represents DA, the other reaction

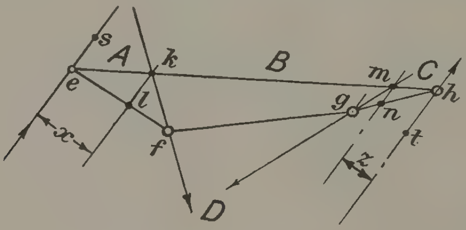

FIG. 88

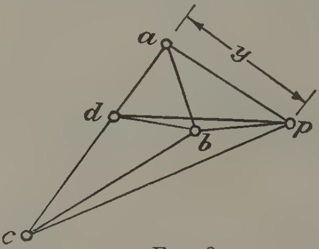

FIG. 89

We have already shown in Lesson VIII that for any point, say, k, the moment of the force DA about $k = kl$, the ordinate of the funicular polygon, measured to scale. Consider the two triangles ekl and adp. The angle at k is equal to the angle at d, and the angle at l is equal to the angle at a; hence the triangles are similar, and we may write—

$$\frac{lk}{x} = \frac{ad}{y}$$

where x = perpendicular from e let fall on lk,
and y = perpendicular from p let fall on ad.

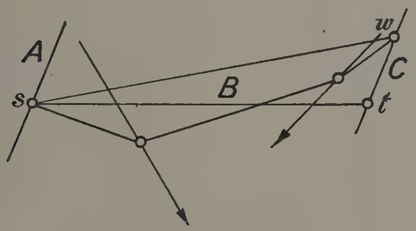

FIG. 90

But lk represents the moment of DA about $k = DA \times x$

$$\therefore \frac{DA \times x}{x} = \frac{ad}{y}$$

or, $ad = y \times DA$ . . . . (1)

Similarly, we may deal with the two triangles hmn and pdc; these are similar triangles, and hence we may write—

$$\frac{mn}{z} = \frac{dc}{y}$$

But mn represents, to scale, the moment of CD about $m = CD \times z$

$$\therefore \frac{CD \times z}{z} = \frac{cd}{y}$$

or, $cd = y \times CD$ . . . (2)

From (1) and (2), we see that

$$\frac{ad}{cd} = \frac{y \times DA}{y \times CD}$$

or $ad : AD :: cd : CD$;

that is, a line drawn through p, and parallel to the closing line, eh, of the funicular polygon, divides the resultant of the loads into two parts which, separately, represent the two reactions.

Again, consider the same system of forces as in Fig. 88. It is now required to draw the funicular polygon which shall pass through the two points s and t. Construct the force diagram, Fig. 91, using *any* trial pole point p. Draw the funicular polygon, commencing at s. It is seen that the funicular polygon does not pass through t, so that clearly, the trial pole point p has been wrongly located. The point d, however, has been correctly determined, since, in the proof given above, the pole point was taken at random. The correct pole *must* lie on a line passing through d, and parallel to the closing line st of the desired funicular polygon. Through d draw dp, parallel to st; then a funicular polygon drawn with *any* pole point on this line will pass through t when started from s, and vice versa.

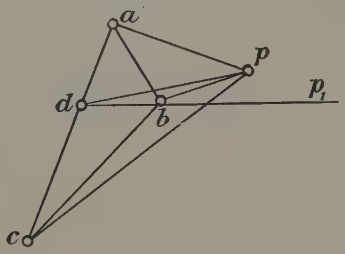

FIG. 91

To Draw a Funicular Polygon Through Three Given Points

An extension of the construction given above enables us to draw a funicular polygon through three given points, s, t, and u, Fig. 92.

Set down the line of loads, a, b, c, d, e, and f, Fig. 93, and draw the radial lines from any pole point, p, to these points. Consider that part of the funicular polygon as being a separate polygon which is to pass through the points s and t. In the space C, Fig. 92, draw ktl parallel to pc; kh parallel to pp, and hg parallel to pa; sg being parallel to ac. Draw pv parallel to gt, the closing line of the funicular polygon. But, the closing line of the funicular polygon is required to be st. From v, draw vp_1 parallel to ts; then, a funicular polygon drawn by means of a pole point situated anywhere on vp_1 will pass through s, when started from t. Similarly we may deal with the funicular polygon $tlmnq$; pr is parallel to tq and rp_1 is parallel to tu. Then, as in the first case, a funicular polygon drawn by means of a pole point situated anywhere on rp_1 will pass through *u*, when started from *t*; p_1 is, therefore, a pole point which may be used to draw a funicular polygon passing through the three given points *s*, *t*, and *u*. In order not to complicate the diagrams, Figs. 92 and 93, the correct funicular polygon has not been drawn, but the reader is urged to test the correctness of position of the pole point p_1 by drawing the funicular polygon for himself.

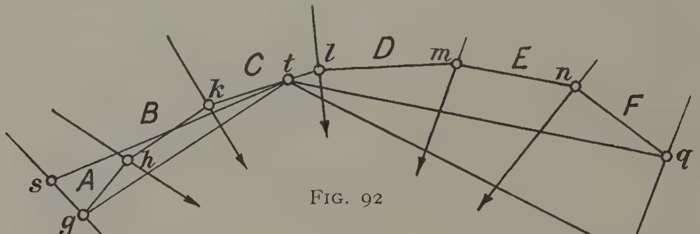

Fig. 92

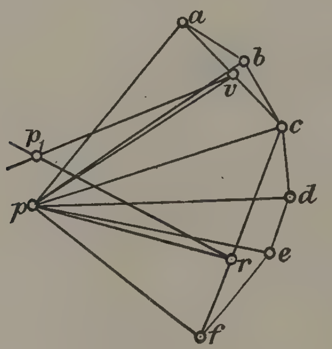

Fig. 93

Three-pinned Circular Arch

An application of the construction given above occurs in a *three-pinned circular arch*. In this structure the joints are assumed to be frictionless, and hence, the sides of the funicular polygon, which in this case we refer to as the *linear arch* or *line of thrust*, must pass through the springings and the crown.

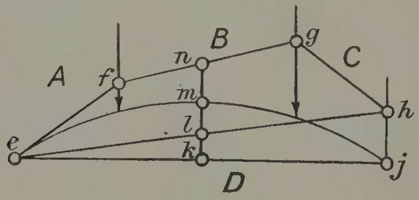

Fig. 94

The centre line of a three-pinned circular arch is shown in Fig. 94. We require to draw the linear arch. Set down the line of loads *abc*, Fig. 95, and choose any trial pole point *p* Commencing at the left-hand pin joint, *e*, draw the *linear arch efgh*. Draw *pd* parallel to *eh* But, *eh*, the closing line of the funicular polygon ought to be horizontal and coincide with *ej*, in order that the funicular polygon shall pass through *j*; draw dp_1 parallel to *ej*. Then, since vertical heights within the closed funicular polygon are inversely proportional to the tria polar distance, we may write—

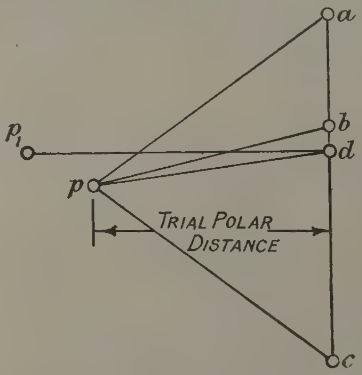

Fig. 95

$$\frac{km}{nl} = \frac{\text{trial polar distance}}{\text{true polar distance}}$$

and hence the true polar distance, dp_1 is obtained

Example. A three-pinned circular arch ha a span of 100 ft. with a rise of 20 ft. The arch carries a load of 5 tons at a point 16 ft. from the left-hand support, and a load of 10 tons at a point 30 ft. from the right-hand support. Draw the linear arch, and state (1) the bending moment and the thrust on the arch under the 10 ton load, (2) the radial shearing force on each side of the section under the 10 ton load.

The arch is shown set down to scale in Fig. 96. Set down to any convenient load scale, say, 1 in. to 2 tons, the line of loads *abc*, Fig. 97. Select, at random, any pole point *p*, and draw the funicular polygon *ijklz*. It is clear, now that *p*, the pole point selected is not the required one, since the sides of the funicular polygon, which now serve as the line of thrust or linear arch, do not pass through *n* and *r*. The point *d* on the line of loads is obtained in the usual way, by drawing *pd* parallel to *il*, Fig. 96. As has been explained in reference to Figs. 94 and 95, draw dp_1 parallel to *iqr*, Fig. 96, and obtain the correct polar distance by proportion—

$$\frac{mz}{nq} = \frac{\text{correct polar distance}}{\text{trial polar distance}}$$

$$\text{or,} \quad \frac{1{\cdot}42 \text{ in.}}{2{\cdot}5 \text{ in.}} = \frac{\text{correct polar distance}}{8{\cdot}5 \text{ in.}}$$

$$\therefore \text{ correct polar distance} = \underline{4{\cdot}83 \text{ in.}}$$

Set off $dp_1 = 4{\cdot}83$ in., join p_1a, p_1b, and p_1c, and, commencing at *i*, draw the required funicular polygon, which will now be found to pass through *n* and *r*. The *line of thrust* or *linear arch* is *iwnsr*. To determine the bending moment under the 10 ton load, we take the moment of R_2 about the point *g* as follows—

Bending moment acting on the arch section under 10 ton load $= R_2 \times gh$.

Bending moment acting on the arch section under 10 ton load $= 12{\cdot}4$ tons $\times (0{\cdot}7 \times 8)$ ft.

Bending moment acting on the arch section under 10 ton load $= 69{\cdot}44$ tons. ft.

The thrust is obtained by resolving R_2 in a direction parallel to the tangent to the arch drawn at *g*. This is conveniently shown on Fig. 97; *cf* represents the thrust on the section of the arch under the 10 ton load = 11·42 tons. Again, the normal component of R_2, which is represented by fp_1 Fig. 97, is the radial shearing force on the section of the arch immediately to the right of the 10 ton load. This shearing force is scaled off the diagram as 4·86 tons. The radial shearing force on the section immediately to the left of the 10 ton load is given by the radial component of the thrust p_1b, Fig. 97, viz., p_1t, and this component is scaled off as 4·76 tons.

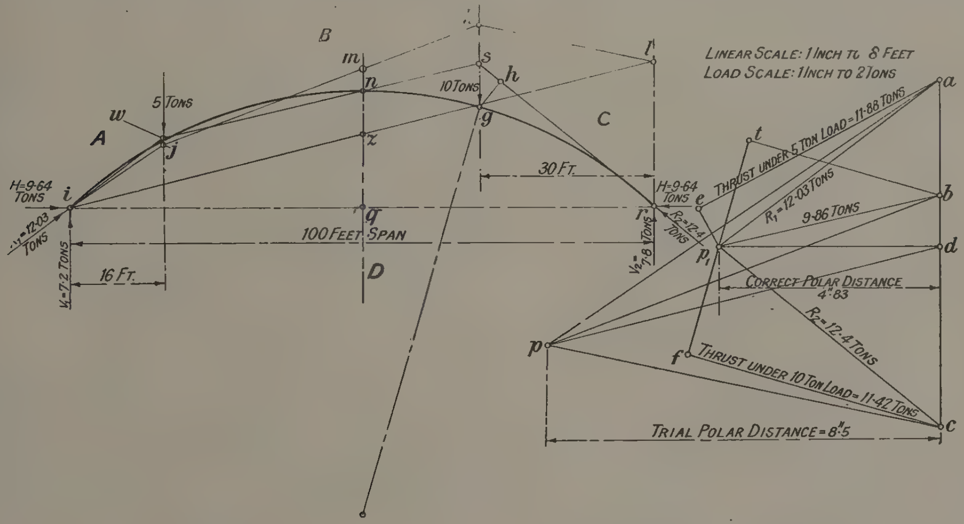

FIG. 96

FIG. 97[1]

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[1] Note: These figures are ½ the size of the original

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
Lecturer in Mathematics, Battersea Polytechnic

LESSON XVII

TRIGONOMETRY—(*contd.*)

Graphs of the Ratios. In the last two lessons we have been thinking of the trigonometrical ratios as numbers contained in tables—very useful numbers—which enable us to perform calculations involving the sides and angles of triangles. Not only right-angled triangles, but any triangles, those not right-angled being divided up into two right-angled ones. Now it must be clear that there are formulae connecting the sides and angles of a triangle, but one difficulty has to be faced before we attempt to find and use these. We have not yet defined the ratios for an angle greater than a right angle, though it is possible that a triangle may contain an obtuse angle; this must be examined. Considering also one of the important examples we have solved, that on the motion of a piston, where the displacement of the piston is calculated from the dimensions of crank and connecting rod, and the angle turned through by the crank, it is clear that there must be, again, a formula for the piston displacement in terms of the angle. But the crank goes on turning—not merely through one, or two, or four right angles, but through *any* number of degrees. To cover this very important case, we need to be able to find the ratios for *any* angle, however great—and even for *negative* angles (the idea of a negative angle should cause no trouble, it merely means an angle measured *the other way* round, clockwise instead of anti-clockwise, or vice versa). We may also add here that this instance of the piston's motion is important in showing the connection of sines and cosines with reciprocating or vibratory motion. In fact, the mathematical expression of such a motion is only possible in terms of sine or cosine. For this reason, a good knowledge of trigonometry is essential in dealing with alternating currents, in electrical engineering, and for *all* cases of vibratory motion.

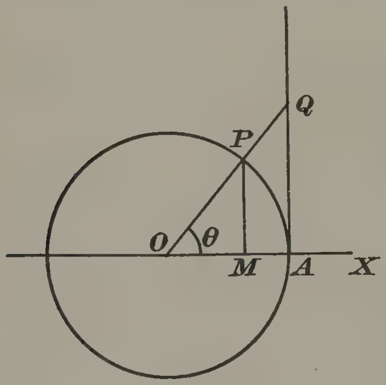

FIG. 67

The extension of the idea of sine, cosine, etc., to angles greater than 90° is very well made through the graphs of these ratios, and the piston motion suggests a connection between a rotating crank and the trigonometrical ratios which we shall follow up. Imagine a crank, OP (Fig. 67), of unit length, to have rotated through an angle XOP from the line OX. Then if PM is perpendicular to OX, and we denote $\underline{/XOP}$ by θ, we shall have

$$PM = \sin\theta, \quad OM = \cos\theta,$$

and by imagining OP to rotate we can see the changes in the values of the sine and cosine. Also, if OX cuts the circle in A, and AQ is drawn perpendicular to OX to meet OP produced in Q, then, also,

$$AQ = \tan\theta,$$

and the changes in the values of the tangent can be seen by imagining OP to rotate.

Sine. We will first follow and graph the changes in the perpendicular PM, as OP rotates, and thus show how to find the *sine* of *any* angle. In Fig. 68, we have shown the crank at every 30°, numbering them 0, 1, 2, 3, 4, etc. We have also marked off the horizontal axis in degrees. We then plot the points on the graph by drawing the dotted horizontals through the points 0, 1, 2, ... on the circle, obtaining the points 0, 1, 2, ... on the graph where these horizontals cut the verticals through 0°, 30°, 60°, ... We get the "wave" graph which we shall learn is the characteristic of sines and cosines. It must be

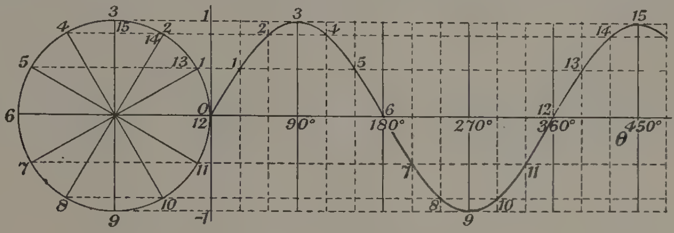

FIG. 68. GRAPH OF SINE

clear that it repeats itself in every revolution—it is shown for a quarter of the second. Examining the graph more in detail, we see that from 0° to 90°, the sine increases from 0 to 1 (as we already know), and this part of the graph could be plotted from a table of sines. Between 90° and 180° the graph falls and the sine *de*creases from 1 to 0 again. Moreover, this part of the graph and the part from 0° to 90° are exactly symmetrical, so that the values of the sine from 90° to 180° repeat, but in the reverse order, those from 0° to 90°. From the method of construction we see, in particular, that

$$\sin 120° = \sin 60°, \ \sin 150° = \sin 30°,$$

and from the graphs that, quite generally,

$$\sin \theta = \sin (180° - \theta).$$

Thus to find the sine of an obtuse angle, look up in tables the sine of the supplementary angle.

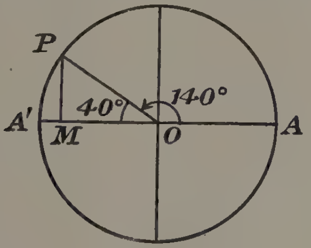

FIG. 69

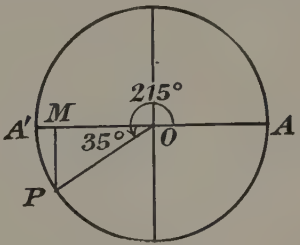

FIG. 70

Beyond 180°, the tracing point is *below*, instead of above, the horizontal diameter. This "other way-ness" is, as usual, taken account of by using the − sign—values below the line are *negative*. Thus, we see that from 180° to 360°, the sine is negative, but the numerical values repeat those from 0° to 180°. Beyond 360° the cycle of values is repeated once every 360°, indefinitely. In particular,

$$\sin 210° = -\sin 30°, \qquad \sin 240° = -\sin 60°$$
$$\sin 270° = -\sin 90° = -1, \ \sin 300° = -\sin 60°,$$
$$\sin 330° = -\sin 30°, \quad \sin 360° = \sin 0° = 0,$$

and so on. Thus, the sine of *any* angle is numerically equal to the sine of some acute angle, and, though its algebraic sign must be determined from the circle diagram or the graph, its value can be found from the ordinary table of sines of angles from 0° to 90°. To find the angle and sign it is a great help to remember the form of the graph, but probably the safest way is to sketch the circle diagram. Draw the circle, putting in the horizontal and vertical radii dividing it into quadrants, put in the radius (crank) at the required angle (a rough sketch is sufficient) and now determine the "corresponding acute angle," and the algebraic sign. As instances—

1. Sin 140°. The circle diagram is drawn in Fig. 69, in which *OP* makes 140° with *OA*. *P* is *above* the horizontal diameter, so the sine is *positive*. The corresponding acute angle, *POA′*, is 180° − 140° = 40°. So

$$\sin 140° = + \sin 40° = \underline{0{\cdot}6428}.$$

2. Sin 215°. Circle diagram in Fig. 70. *OP* making an angle of 215° in the anti-clockwise direction with *OA*. *P* is *below* the horizontal,

so the sine is *negative.* The corresponding acute angle is $\underline{/POA'}$, $215° - 180° = 35°$. So

$$\sin 215° = -\sin 35° = -0{\cdot}5736.$$

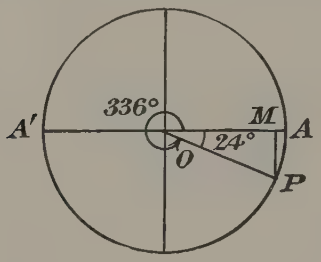

Fig. 71

3. Sin 336°. The circle diagram is Fig. 71, OP making an angle 336° in the anti-clockwise direction with OA. P is *below* the horizontal, so the sine is *negative.* The corresponding acute angle, $\underline{/POA}$, is $360° - 336° = 24°$. So

$$\sin 336° = -\sin 24° = -\underline{0{\cdot}4067.}$$

4. Sin 941°. If OP turns through 941° from OA, it turns through two complete revolutions (720°) and then 221° more, its final position being as in Fig. 72. Thus, firstly, sin 941° = sin 221°. By the methods above, we see that the sine is *negative,* and that the corresponding acute angle is 41°. So

$$\sin 941° = -\sin 41° = -\underline{0{\cdot}6561.}$$

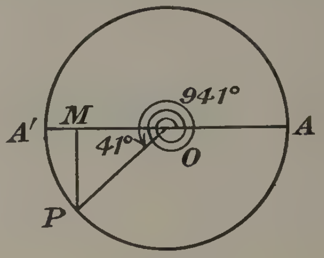

Fig. 72

From these examples it should be clear that the "corresponding acute angle" is the difference between the given angle and the nearest multiple of 180° (in ex. 4, for instance, the nearest multiple of 180° to 941° is 900°; the difference between 941° and 900° is 41°, the acute angle used). After a very little practice, the algebraic sign should cause no trouble.

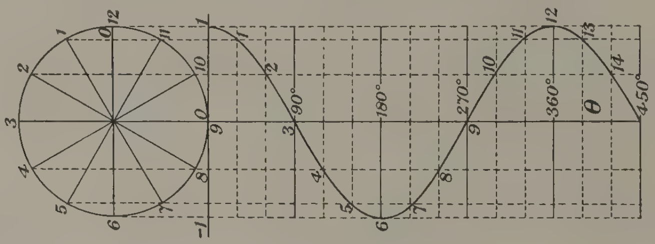

Fig. 73. Graph of Cosine

Cosine. To deal with the cosine in a similar way, we must graph the changes of value of OM (Fig. 67). The most obvious way to do this is to plot the angles along a vertical axis, drawing verticals through the points on the circle to meet the horizontals through the corresponding points on the axis. The graph so constructed is shown in Fig. 73. In this graph we see, as we already know, that the cosine decreases from 1 to 0 as the angle increases from 0° to 90°. Beyond 90° the tracing point is to the *left* instead of to the *right,* of the vertical, and consequently the cosine is now *negative.* From 90° to 180° the cosine goes from 0 to −1, then, from 180° to 270° retraces its steps back to 0, and then, from 270° to 360° increases from 0 to 1. Beyond 360° the cycle is repeated indefinitely. It will be seen that the values of cosines of angles greater than 90° are all obtainable, numerically, as cosines of the corresponding acute angles, these corresponding acute angles being found in exactly the same way as for the sine. In particular, the cosine of an obtuse angle is negative, numerically equal to the cosine of the supplementary angle. That is,

$$\cos \theta = -\cos (180° - \theta).$$

We may find the cosines of the angles for which we have already found the sines above.

1. Cos 140°. Equivalent acute angle, 40°; P to *left* of vertical (see Fig. 69), so that the cosine is *negative.* So

$$\cos 140° = -\cos 40° = -\underline{0{\cdot}7660.}$$

2. Cos 215°. Equivalent acute angle, 35°; *P* to *left* of vertical, so that the cosine is *negative* (see Fig. 70). So

$$\cos 215° = -\cos 35° = -\underline{0{\cdot}8192}.$$

3. Cos 336°. Equivalent acute angle, 24°; *P* to *right* of vertical, so cosine is *positive* (see Fig. 71). So

$$\cos 336° = +\cos 24° = \underline{0{\cdot}9136}.$$

4. Cos 941°. Equivalent acute angle, 41°; *P* to *left* of vertical, so cosine is *negative* (see Fig. 72). So

$$\cos 941° = -\cos 41° = -\underline{0{\cdot}7547}.$$

It will be seen from Fig. 73 that the shape of the cosine graph is exactly the same as that of the sine graph, only starting from a different point. The sine is always 90° behind the cosine, as can be seen in Fig. 74, in which both graphs are drawn on the same axis. They have both been continued back some distance, to show the sine and cosine of negative angles.

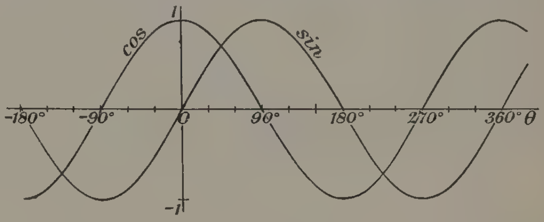

FIG. 74

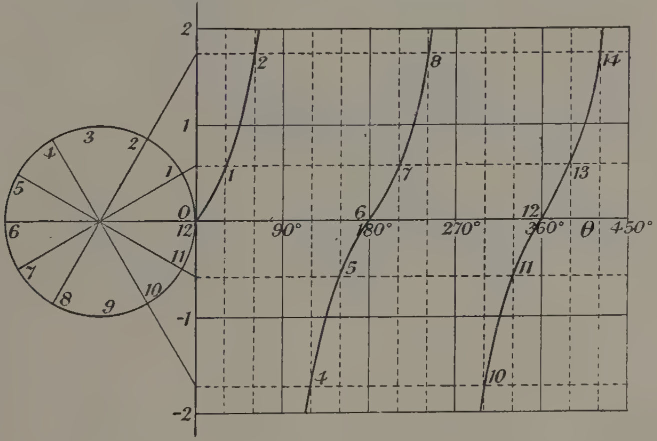

FIG. 75. GRAPH OF TANGENT

Tangent. To obtain the graph of the tangent, we must study the changes in the value of AQ (Fig. 67). This is done in Fig. 75, by continuing the radii until they meet the vertical tangent, and drawing horizontal lines through the points of intersection, points on the graph being found at the intersections of these with the verticals through the corresponding points of division of the horizontal axis. From 0° to 90° we see the tangent increasing, and becoming infinite at 90°. Beyond 90° the radius has to be produced *backwards* to meet the vertical tangent to the circle, and it meets it *below* the horizontal axis. From 90° to 180°, therefore, the tangent is *negative*, increasing algebraically from $-\infty$ to 0. Beyond 180°, the radii produced backwards meet the vertical tangent *above* the horizontal axis, so the tangent is again *positive*. From 270° to 360°, the radius produced meets the vertical tangent *below* the horizontal axis, so the tangent is again *negative*. The graph then repeats itself indefinitely—though, as a matter of fact, it repeats itself every 180°. To find the tangents of angles greater than 90°, we have again to find, by exactly the same method, the corresponding acute angle, and also to determine the sign. This sign is positive or negative for alternate quadrants, a fact worth knowing, but it is probably better to determine the sign from that of the sine and cosine, for, as can easily be verified, both in sign, as well as in value, the relation found in the last lesson,

$$\tan \theta = \frac{\sin \theta}{\cos \theta}$$

holds for *all* values of θ. Thus, in particular instances,

1. tan 140°. Sin +, cos −; therefore, tan is −

$$\tan 140° = -\tan 40° = -\underline{0{\cdot}8391}.$$

2. Tan 215°. Sin –, cos – ; therefore, tan is +
tan 215° = + tan 35° = 0·7002.

3. Tan 336°. Sin –, cos + ; therefore, tan is –
tan 336° = – tan 24° = – 0·4452.

and so on.

Other Ratios. The other ratios are reciprocals of sine, cosine, and tangent, so there should be no difficulty in finding the signs to be attached to them for any angle. There is, however, no simple way of drawing the graphs geometrically.

EXERCISE No. 28

Find the values of—
sin 135°; cos 165°; tan 240°; cos 290°; sec 210°; sin 440°; tan 941°.

Answers to Exercises 26 and 27

(26)

1. 56·94 yd. = 170·8 ft.
2. 13° 36′.
3. 10·82 miles W., 33·29 miles N.
4. 6·883 ft., 19·66 ft.
5. 57° 54′, 122° 6′, 3:873 in.
6. (*a*) 7·30 lb., 18° 21′ with the 5 lb. force.
(*b*) 8·718 tons, 23° 26′ with the 10 ton force.
7. 18° 12′ W. of N., 79·22 m.p.h.
8. 14° 20′ E. of N., 72·12 m.p.h.

9.

Angle	0°	15°	30°	45°	60°	75°
Displacement (in.)	0	0·366	1·431	3·089	5·180	7·518

Angle	90°	105°	120°	135°	150°
Displacement ,,	9·909	12·175	14·179	15·817	17·019

Angle	165°	180°
Displacement ,,	17·753	18·000

10. 53° 8′; 59° 29′; 67° 23′.

(27)

1.	sin	cos	tan	cot	sec	cosec
(*a*)	0·4359	0·9000	0·4843	2·0648	1·1111	2·2941
(*b*)	0·9847	0·3162	3·0000	0·3333	3·1623	1·0541
(*c*)	0·3049	0·9524	0·3201	3·1240	1·0500	3·2798

2. s, $\sqrt{1-s^2}$; $\frac{s}{\sqrt{1-s^2}}$; $\frac{\sqrt{1-s^2}}{s}$; $\frac{1}{\sqrt{1-s^2}}$; $\frac{1}{s}$.

3. 0·5724; 0·82.

MACHINE TOOLS

By Joseph G. Horner, A.M.I.Mech.E.

LESSON XV

THE SHAPING AND SLOTTING MACHINES

The Shapers

Characteristic Features. In these, the tool is carried at the end of a horizontal ram, which reciprocates over the work that is bolted to a knee which has vertical adjustments on the face of the main framing. Designed by Nasmyth for dealing with short articles only, the travel of the ram is still limited by inches, rarely exceeding 24 in. Within these limits it is a useful machine. In the smaller, pillar machines, the width tooled is limited to the lateral traverse or feed of the table, but this can be increased to several feet by mounting the ram in a head which is traversed along a bed—the travelling head shapers. Its principal value lies in the tooling of small articles that can be fixed up and presented to the tool to deal with shapes that cannot be dealt with on the planer. On the other hand, if similar pieces are required in large numbers, and the cutting is of a straightforward character, these should be done in tandem on the planer table.

Much variation exists in the designs of shapers. A familiar type is shown by Fig. 102, by the Queen City Machine Tool Company, of Cincinnati, Ohio, U.S.A. The work carrying knee has adjustments, both horizontally and vertically, on a saddle that slides on vertical ways on the pillar. The tool, clamped in a holder, generally resembling that on a planer, reciprocates over it and is fed downwards. Lateral feed is imparted to the work on the knee along its saddle in the pillar machines, but to the tool saddle in the traversing head shapers. These machines may have one tool ram and one knee, or one tool arm and two knees for the accommodation of long pieces. Double head machines have two tool arms, each with its knee, mounted on the same bed, and each traversed independently of the other.

Quick Return. In the smaller machines, say, those of less than about 8 in. stroke, the speeds of cutting and return of the tool ram are equal. But in all those of longer range the Whitworth quick return, or something equivalent to it, is fitted. In this, Fig. 103, a pinion *A* rotates a toothed wheel *B*, and carries with it a die block *C*, that slides in a link *D*, pivoted at *a*. The end of the link farthest from the pivot is pivoted to a connecting link *E*, the opposite end of which is attached to the tool ram. The result is, that when the die block is in the position of the largest radius in the pivoted link it is moving slowly and controlling the cutting stroke. When in the smallest radius it is returning the ram quickly. An objection to the movement is that the rate of cutting is not quite uniform. Variations are made in the ratios by adjusting the block radially in its link with a screw in the wheel, which also alters the length of stroke of the ram. Two other methods are in use, but they are not so simple, and the Whitworth design is mostly retained. It has the great advantage that it is precise, so that cutting can be done to a line. In one design, Fig. 104, the quick return is obtained by a crossed belt from a pulley on the countershaft that is larger than the driving pulleys. There are two of these, giving two cutting speeds. A friction clutch, always running, lying within the rims puts either into motion. The drive from the pulley shaft for cutting and reverse goes through spur gears to the bull wheel beneath the ram. In this way uniform speeds are secured through the length of cut.

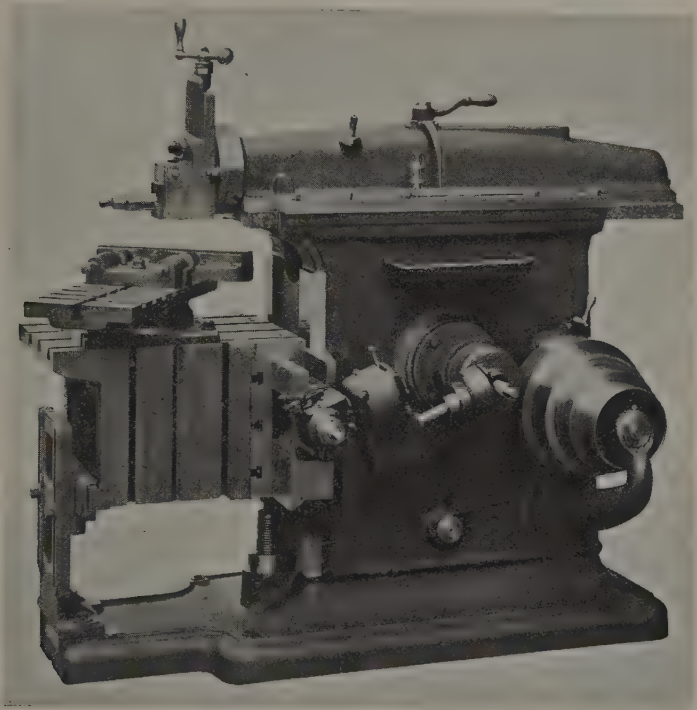
Fig. 102. Typical Shaping Machine

An objection to all the older designs is that the adjustment of the block in the rocking link involves stopping the machine. In many of the later models this is regulated from the outside, while the machine is running. In the shaper shown in Fig. 102, the screw that imparts the radial adjustments to the block is actuated through mitre gears from outside by the turning of a shaft that goes through an extended boss of the crank wheel. In another machine, the pushing inwards of a hand wheel outside the pillar engages gears that actuate a bevel pinion on the end of the screw which adjusts the radial position of the die block. A spring ball catch locks the mechanism when the adjustment has been made.

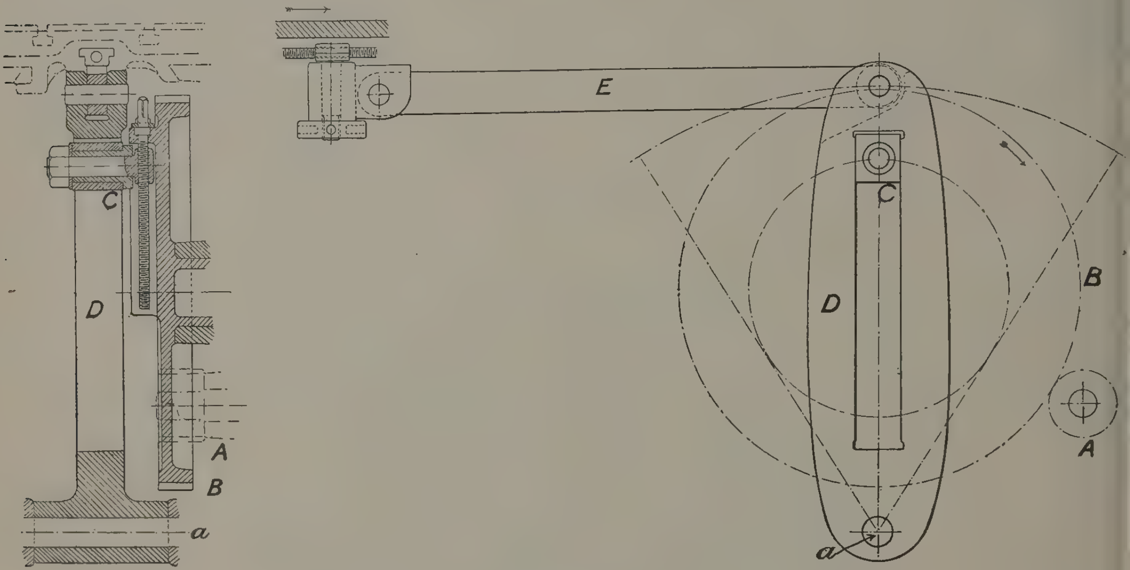

Fig. 103. Quick Return Motion of Shaper

Position of Stroke. This is varied by shifting the position of the termination of the connecting link in the ram. It is effected by the clamping handle seen at the top of the ram in Fig. 102. This locates and controls the action of the tool over that portion of the work that has to be shaped, while the stroke adjustment changes the amount of travel of the tool.

Speeding and Feeding. The speeds of the ram are still obtained in most cases with belt cones, with three or four steps. These in the larger machines are supplemented with back gears, so doubling the speeds. Some later machines have a single belt drive with a gearbox. In others, a motor drive with a constant speed includes geared changes. Or a variable speed motor, clutch connected, transmits speeds through a belt, or a chain. Feeds are derived from the speed pulley, geared down through ratchet wheels, and actuating a screw that runs longitudinally. A short shaft that carries the disc for quick return receives a slotted feed disc. Thus, for each speed of the stepped belt cone, several rates of feed are available, often as many as sixteen. These control the lateral movements of the work table in the pillar machines, but those of the traversing head in the bed type of machines. In all the smaller machines, the down feed of the tool box is effected by turning a crank handle for the screw. Only in the

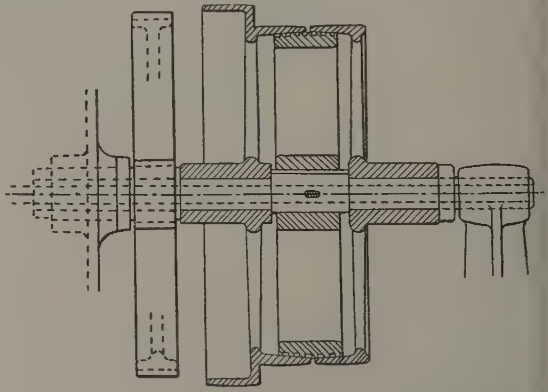

Fig. 104. Pulley and Clutch Drive

larger machines a ratchet and pawl are fitted to feed automatically from the motion of the ram.

Varied Toolings. Valuable features of this machine are the shaping of concave, and convex surfaces. The first is performed with an arc movement imparted to the tool box with worm and quadrant gear. This is seen in Fig. 105. The second is effected by mounting the work on an arbor which is rotated through an arc, or a complete circle, Fig. 106, by means of worm gear actuated with a ratchet and pawl from the slotted disc that gives feed adjustments. Arbors are made of different diameters and lengths, all interchangeable in their socket, and the work is tightened between cones.

Many variations are apparent in the designs of the later knees and tables. The front face of the pillar or bed, receives a saddle, Figs. 102, 105, 106, on which the knee is fitted. Its section is that of an angle plate to receive work bolted on top, on one or on both sides. It is tee-grooved for bolts. The saddle is elevated by hand, with a crank, bevel wheels, and telescopic screw. The box form of table is common, giving three sides for bolting work to. It is sometimes made to rotate, so presenting faces at an angle. A top is made to tilt to give angles in the other direction, Fig. 106. The table is divided into two portions, a saddle that has vertical movements on the vertical slide faces of the transverse saddle, to which it is aligned with a tongue, and secured with four bolts through a foot. The work table is hinged to the saddle, and its sides embrace two projections with quadrant slots, the curves of which are struck from the centre of the pivot on which the table is hinged, and through which bolts lock the table. In many designs now, the base of the framing is extended as far out as the table, and a steady supports the free end of the table, being adjustable to accommodate itself to different heights of the table, and sliding longitudinally with the feed, Figs. 102, 105. When the knee is removed, deep articles can be bolted on the extension of the base. An essential adjunct to the shaper is the machine vice, Figs. 102, 105. It has jaws with or without swivel. Its base is graduated in degrees to facilitate the setting of work. It is indispensable when small articles have to be set in different angular relations, and for numerous pieces

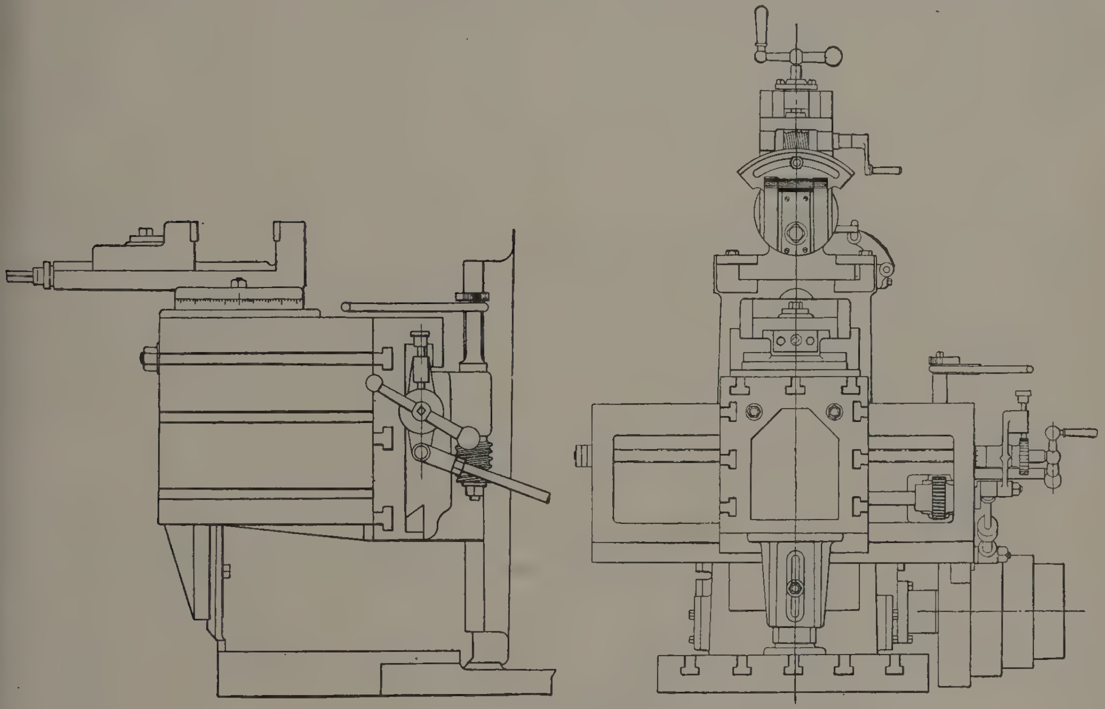

Fig. 105. Details of Shaper Design

that cannot be held by the ordinary methods of clamping down on the table without involving much trouble. Although the shaper has been hard hit by the planer, and more especially by the milling machines and the surface grinders, it survives by reason of the improvements that are embodied in the later designs.

THE SLOTTING MACHINES

A slotting machine may be regarded as a shaper set on end, with the tool ram reciprocating in the vertical direction, the cutting being done on the down stroke, the work lying on the horizontal face of the table. Adjustments and feeds are imparted to the slides of the table, including generally, a self-acting circular motion for the tooling of curved surfaces. The machine is admirably adapted to receive large masses that overhang the table.

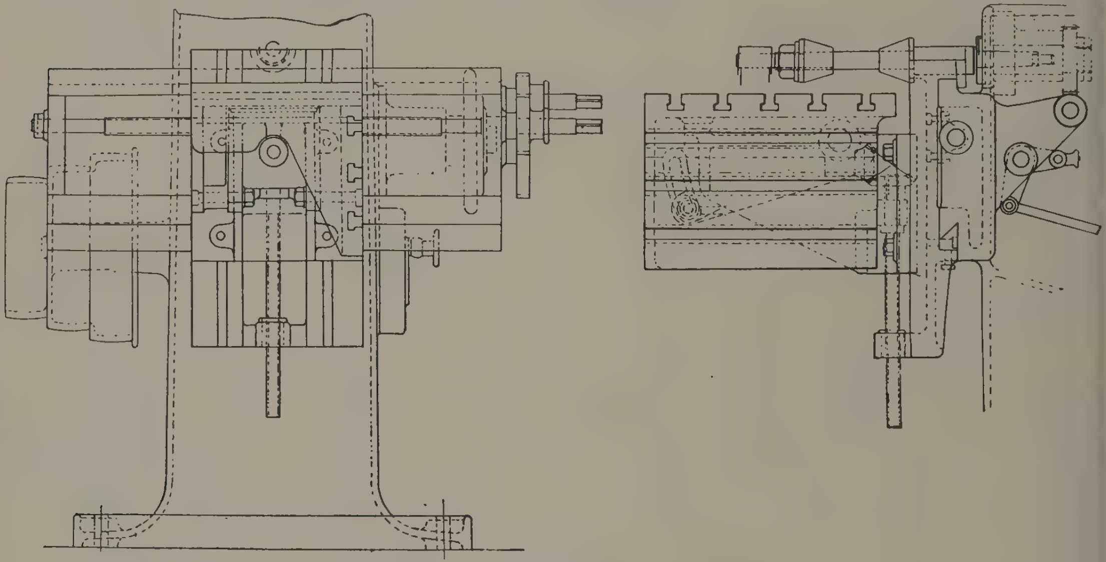

FIG. 106. SHOWING ROTARY ARBOR, AND TILTING KNEE

The Mechanisms. Subject to many variations in the working out of details, Fig. 107 shows the general scheme in a high-class, fully equipped machine. A main frame, in one casting usually, but made in two parts in the largest machines for convenience of transit, includes the horizontal bed on which the table carriage is adjusted towards and away from the tool, and the upright that receives the tool ram and the driving mechanism. The ram reciprocates rigidly in guides with provision for taking up wear. It is counterbalanced with a pivoted weight that rocks with the movements of the ram. Machines have a range up to about 24 in stroke. For all above about 6 in., quick return is provided. This is commonly the Whitwort slotted disc, die block and link connected to th ram, but in some machines elliptical gears perform the function, and in a few, pulleys of differen diameters for cutting and return. Spee changes are effected with stepped belt pulleys or frequently now with a pulley running at constant speed and a box of gears giving from four to six rates. The positions of the ram ar adjusted from the front.

Feeds are derived from a disc on the mai shaft within the framing. It has a heart shaped cam cut on one face which actuates rocking lever that descends to the lower par of the framing, where it controls a pawl tha engages intermittently with a ratchet wheel o the end of a horizontal shaft that actuates gears at the opposite end, and screws that move th table slides. These are, the saddle inwards an outwards, the cross slide giving transvers movements and the circular feed to the to table through worm gears. These, actuate automatically, also have hand adjustments fo setting work. The upper table is sometime hinged to permit of angular slotting. In a fe machines, angular adjustments are imparted t the head that carries the ram. Most tables hav a suds tray surrounding them.

The slotter has been hit more hardly perhap than any other machine tool by better designs

Its weak feature is the overhang of the tool from the ram in deep cutting, which causes deflection to occur. The slotting of key grooves, long its principal function, has been taken over by the key grooving and broaching machines, and the shaping of curved outlines, as the ends of connecting rods is as frequently done in vertical spindle milling machines. But a large volume of work remains. A very special design is the frame plate machine, for slotting the edges of locomotive frame plates laid in a pile.

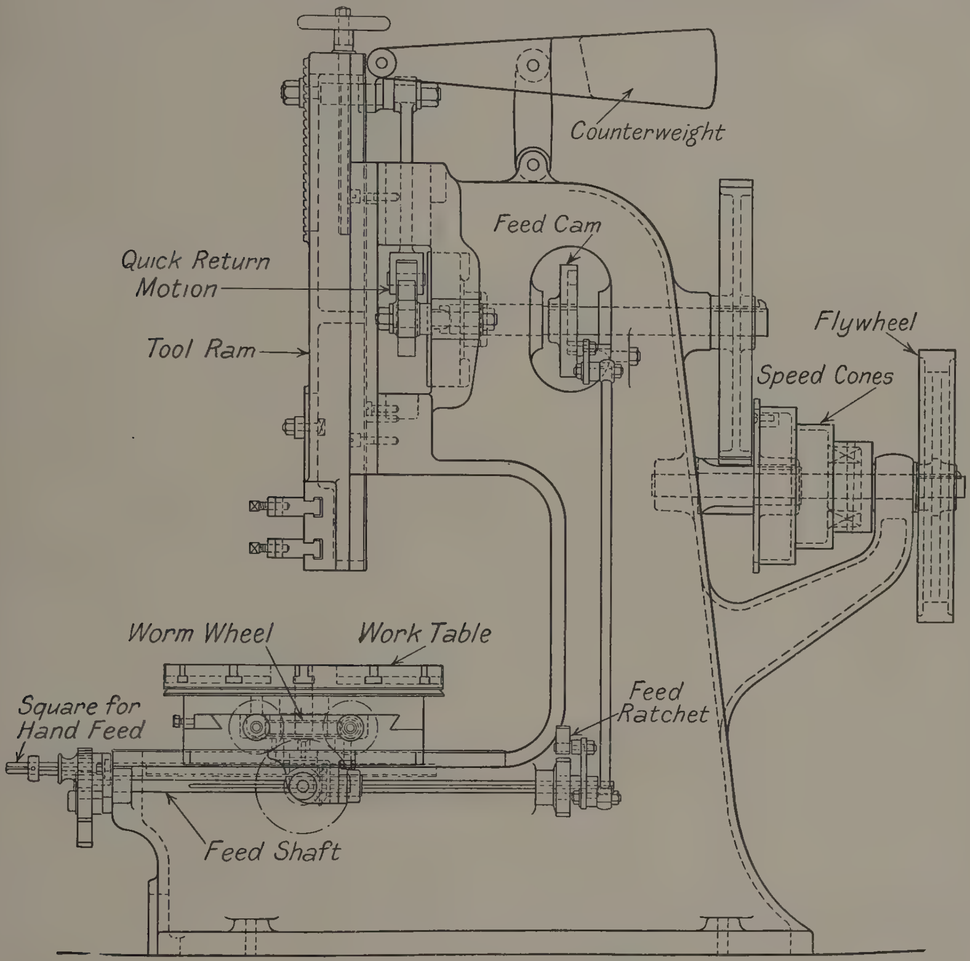

FIG. 107. SLOTTING MACHINE

ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

By A. P. Young, O.B.E., M.I.E.E.

LESSON VIII

THE PRODUCTION OF ELECTRICITY FROM MAGNETISM

It was explained in the previous lesson that whenever current flows in a conductor, a magnetic field is produced. Conversely, it is also true that a magnetic field can be made to produce an electric current in a conducting circuit. In fact, electric and magnetic effects are inseparable, and one effect cannot be produced without the other being linked with it. This fundamental discovery underlying the production of electricity from magnetism—or as it is termed *electro-magnetic induction*—was made by Faraday in the autumn of 1831.

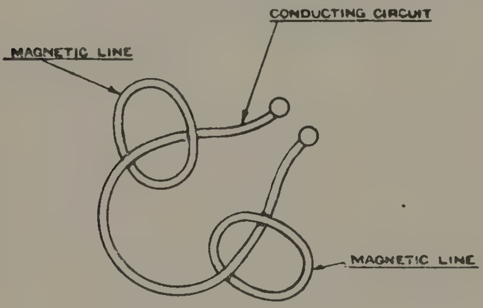

Fig. 40. Diagram Illustrating Linkage

1. **Faraday's Experiments.** The experiments of Faraday on electro-magnetic induction may appear simple and commonplace in this age of electricity, but it must be remembered that 12 years elapsed, after Oersted's experiment, before a method of producing electricity from magnetism was discovered. It was reserved for Faraday to show the world how this result could be achieved. It is of interest to record that Faraday worked for seven years on the problem that he set himself to solve, of converting magnetism into electricity, before success was attained. The principle of electromagnetic induction evolved by Faraday as a result of his many beautiful experiments, is the fundamental basis of the design of all our present-day electrical generators, alternators, transformers, and in fact, all electrical apparatus now employed for commercially converting mechanical energy into electrical energy. The discovery was an epoch-making one in that it laid the foundation of the electrical industry which is even now still in its infancy.

These experiments proved that there must be relative movement between a magnetic field and a conductor, in order to induce in the conductor an E.M.F. which will give rise to a current if the circuit be closed. In other words, *a current is always induced or created in a closed electric circuit, if and when, the number of magnetic lines linked with that circuit is changed*

It is well that the student should clearly understand what is meant by "linkage." In Fig. 40 is depicted a single turn of an electric circuit *linked* with imaginary magnetic lines. In order to change the number of magnetic lines of force linked with such a circuit, either by increasing or decreasing this number, the conductor must necessarily cut through the magnetic lines. Hence the familiar electrical phrase —*cutting magnetic lines of force.*

Faraday's experiments on electro-magnetic induction are illustrated in Fig. 41. He found that—

1. When a bar magnet was plunged into a coil, a deflection was obtained on the galvanometer indicating the presence of an induced current. Likewise, on removing the magnet from the coil, a deflection was obtained—but in the opposite direction.

2. In the second experiment, separating the magnet poles resulted in a deflection of the galvanometer, whilst on bringing them together again the galvanometer deflected in the opposite direction.

3. With two coils having entirely separate circuits and in no way connected electrically with one another, the closing or opening of the battery switch in the circuit of one coil (primary) was accompanied by a galvanometer deflection indicating an induced current in the other coil (secondary), the deflections for opening and closing the switch being in opposite directions.

4. The top diagram is a further development of 3, the coils being wound in this case on a ring of soft iron wires. The number of magnetic lines induced as a result of current flowing in

the primary coil will be very much greater owing to the better magnetic path. The results of this experiment were precisely the same as in experiment 3.

This soft iron ring, with its primary and secondary windings, was the very first electric transformer ever made, and is still preserved in the Museum at the Royal Institution in Albemarle Street, London, where Faraday carried out his many researches.

With Faraday's classical ring transformer experiment, the current in the primary winding increases from zero to a maximum steady value (as determined by Ohm's Law) so soon as the switch is closed. During the very small interval of time taken for the growth of the current, the magnetic lines associated with this current are constrained to flow around the iron ring, and in so doing, are forcibly linked with the secondary winding. Thus, a rapid change occurs in the number of magnetic lines linked with the secondary as the flux grows from zero to a steady maximum. In consequence of this rapid change in the flux linked with the secondary, an E.M.F. is induced in this winding.

On opening the switch in the primary circuit, the magnetic lines linked with the secondary windings are reduced to zero. The collapse of the flux linked with the secondary induces an E.M.F. in the opposite sense to that induced on closing the switch. The important point to remember is that an E.M.F. is only induced *so long as the number of magnetic lines linked with the winding is changing.* This change may be effected in many ways so long as there is relative movement between the conductors and a magnetic field.

2. **Law of Electro-magnetic Induction.** The law of electro-magnetic induction, affirmed by Faraday many years after his discovery, states that the induced E.M.F. *is directly proportional to the rate at which the number of magnetic lines linked with the circuit changes.* Expressed in another way, the induced E.M.F. is directly proportional to the *rate of cutting of magnetic lines of force.* The absolute unit of E.M.F. is that induced in a conductor, which moves through a magnetic field at such a rate, that one line of force is cut per second. This, however, is a very small unit which is much too small for practical purposes, so a practical unit called a "Volt" is adopted. The practical unit is 100 million times the absolute unit of E.M.F., i.e.

$$1 \text{ volt} = 10^8 \text{ absolute units} \quad . \quad . \quad (18)$$

Taking the case of the coil and magnet shown in the lower diagram of Fig. 41, if we suppose the coil to have a 1,000 turns and the number of magnetic lines emanating from the S. pole to be 10,000, an E.M.F. of 1 volt will be induced, if the time taken to thrust the S. pole into the coil is one-tenth of a second. In order to induce an E.M.F. of 1 volt, we must, therefore, have a *line-turn* cutting or linkage at the rate of 100 million (10^8) per second. The induced E.M.F. is directly proportional to the flux and the number of turns, and inversely proportional to the time. Expressed mathematically, we have—

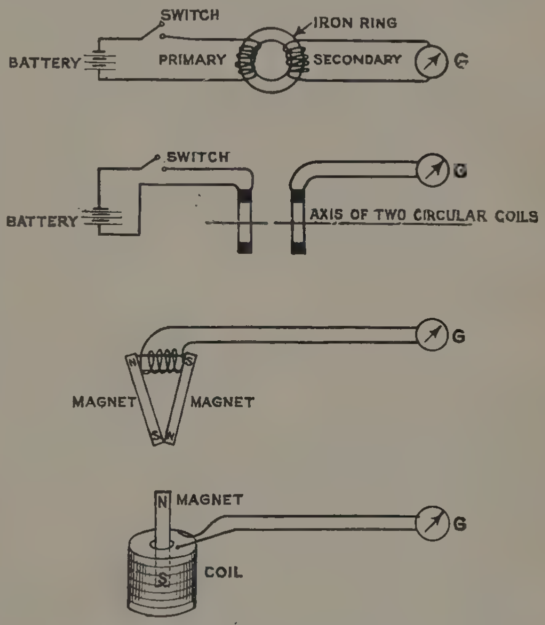

FIG. 41. FARADAY'S EXPERIMENTS ON THE INDUCTION OF CURRENTS

$$\text{Induced E.M.F.} = \frac{\Phi \times S}{t \times 10^8} \text{ volts} \quad . \quad . \quad (19)$$

Where Φ = total number of magnetic lines
S = total number of turns, and
t = time in seconds.

3. **Fleming's Right-hand Rule.** It has already been explained, in dealing with the experiments of Faraday, that by reversing the direction of relative motion between the magnetic field and the conductors, a current in the opposite direction is obtained, as indicated by the reverse deflection of the galvanometer. As

an aid to memorizing the relative directions of magnetic field, motion, and induced current, Fleming's right-hand rule is of value. Referring to Fig. 42, the right-hand diagram shows the relative directions of motion between a magnetic field and a conductor, whilst the left-hand diagram illustrates the application of the rule in determining the direction of the induced E.M.F. or current. If the thumb of the right-hand is pointed in the direction of motion of the conductor, and the index finger in the direction of the magnetic field, then the second finger bent at right angles to the palm of the hand, will indicate the direction of the induced E.M.F. or current.

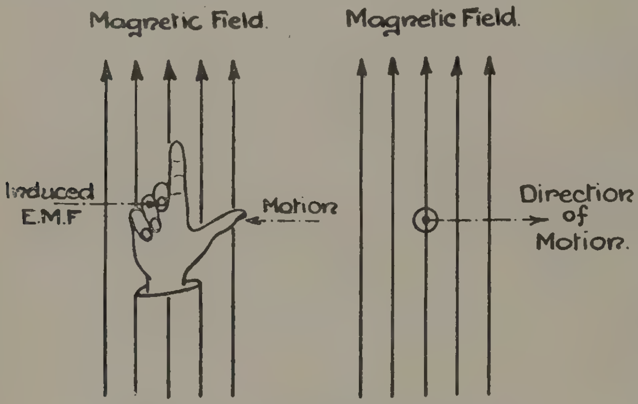

FIG. 42. CONDUCTOR MOVING ACROSS A MAGNETIC FIELD

4. **Graphical Method of Determining Induced E.M.F.** Graphical representations are given in Fig. 43 of the flux and induced voltage changes, in a coil. From the flux curve we can determine the value of the induced E.M.F. at any moment and, by plotting values for successive positions, a curve showing the variation in induced E.M.F. can be obtained. This may be done by determining the *instantaneous rate* of change of the flux, for a number of points on the flux curve, from the equation (19) rewritten–

$$\text{Induced E.M.F.} = E = -\frac{S}{10^8}\left(\frac{d\Phi}{dt}\right) \text{ volts} \quad . \quad (20)$$

where S = total number of turns.

The expression $\left(\frac{d\Phi}{dt}\right)$ simply denotes the instantaneous rate of flux change in respect to time, which can be deduced from the flux curve by drawing a tangent to the curve at any desired point, and determining the tangent of the angle that this line makes with the horizontal.

Referring to Fig. 43, the induced E.M.F. after T seconds, when the flux has reached a value AB, is proportional to $\frac{AB}{BC}$, the tangent of the angle θ. By repeating this calculation for a number of points on the curve, we can, from the values so obtained, plot another curve connecting $\frac{AB}{BC}$ and time, showing how the induced E.M.F. varies with the changing flux in the coil. Such a curve is roughly indicated by the chain-dotted line curve. It should be noted that for an increasing flux, the induced E.M.F. is *negative*, whilst for a decreasing flux, it is *positive*. This explains the meaning of the minus sign in equation (20), because the slope of the curve for decreasing flux is *negative* and the induced E.M.F. therefore *positive*.

5. **Self-induction.** Self-induction is the *inertia* of the electrical circuit, and is exactly analogous to the inertia in a mechanical system.

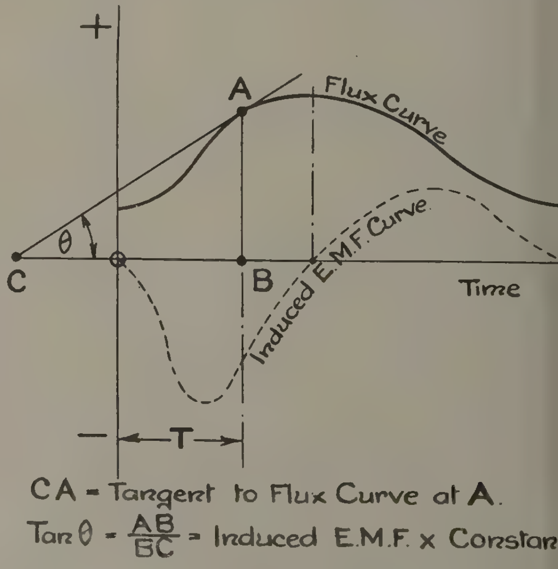

FIG. 43. FLUX AND INDUCED VOLTAGE CURVES

Just as the inertia of a fly-wheel resists any change in its motion, so also does self-induction resist any change in the flow of current in an electrical circuit. When a current is made to flow in a coil, the magnetic lines so induced interlink with the turns of the coil, setting up a reactive, or back E.M.F., which opposes the growth of the current. Likewise, when the current is stopped, the back E.M.F., due to self-induction, acts in such a direction as to oppose the change, thereby tending to maintain the flow of current.

The unit of self-induction is the henry. This is the self-induction of a coil, in which an E.M.F. of 1 volt is self-induced when the current in the coil is changing at the rate of 1 amp. per second; i.e. when a current of 1 amp., flowing in the coil, will produce a magnetic field sufficiently intense to give a *line-turn* linkage of 10^8 per second. The self-induction is denoted by the symbol L.

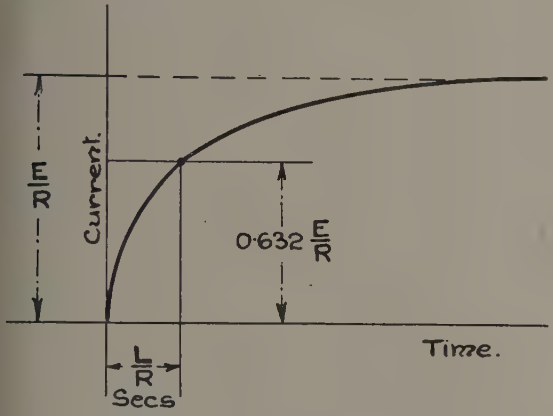

FIG. 44. CURRENT GROWTH IN AN INDUCTIVE CIRCUIT

In an electrical circuit possessing self-induction, the value of self-induced voltage is determined by multiplying the coefficient of self-induction of the circuit by the rate of change of current, and is given by the formula—

$$\text{Self-induced voltage} = -L\left(\frac{di}{dt}\right) \text{ volts} \quad . \quad . \quad . \quad . \quad . \quad (21)$$

Where L = self-induction in henries, and $\left(\frac{di}{dt}\right)$ = rate of change of current, in amperes per second.

In an inductive circuit, the E.M.F., due to self-induction, is of considerable practical importance. Consider, for example, the simple case of a coil of wire suddenly connected to a battery. The effect of the self-induction of the coil is to retard the growth of the current, and consequently, instead of the current increasing instantly to the value determined by Ohm's Law, it rises gradually and the value $\left(\frac{E}{R}\right)$ is finally reached after the lapse of a certain time interval. The rate of increase in current is represented by the exponential curve shown in Fig. 44, and the law of this curve first determined by Helmholtz is—

$$i = \frac{E}{R}\left(1 - e^{\frac{-Rt}{L}}\right) \quad . \quad . \quad . \quad . \quad (22)$$

Where,

i = current in amperes.
E = applied E.M.F. in volts.
R = resistance in ohms.
L = coefficient of self-induction in henries.
t = time in seconds after E.M.F. is first applied.
e = base of naperian logs. = 2·71828.

It is important to remember that the ratio L/R is called the *time constant* of the circuit, and is the time in seconds taken for the current to reach 63·2 per cent of its final and steady value E/R. An increase in the value of L tends to increase the *time constant* of the circuit, whilst an increase in R tends to reduce it.

The self-induced voltage on opening a highly-inductive circuit will, as previously explained, tend to maintain the current flowing in the circuit, and unless precautions are taken, the current will persist in the form of an arc, when the switch contacts separate. Also, this voltage may even reach such a high value as to impair the insulation of the windings. It is the practice, in such cases, to provide an alternative non-inductive path for the current, whilst opening the circuit. Fig. 45 illustrates such an arrangement for breaking the field circuit of a generator. Before the switch actually breaks

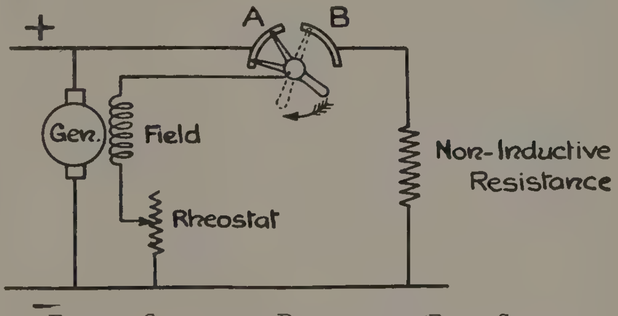

FIG. 45. CONNECTION DIAGRAM FOR FIELD SWITCH

contact with segment A, it makes contact with the segment B, connected to a non-inductive resistance, so that the current in the field coils is allowed gradually to die away to zero.

6. **Lenz's Law.** The opposing or inertia effect of self-induction is expressed in a general law, formulated by Lenz in 1834, and reading as follows—

"In all cases of electro-magnetic induction the induced currents have such a direction that their reaction tends to stop the motion which produces them."

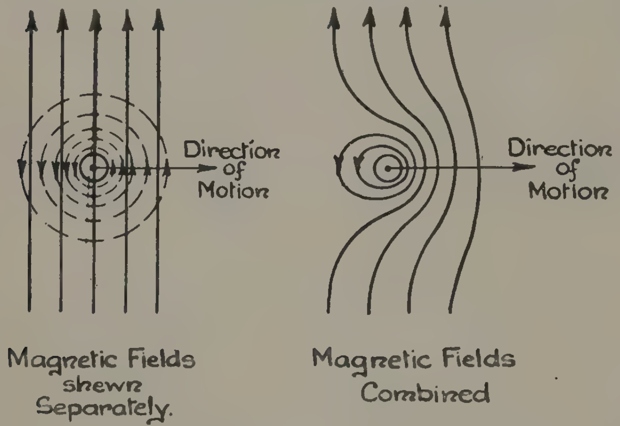

Fig. 46. Diagrams Showing Interaction of Induced Current

The law is illustrated by the case of a conductor (as on the armature of a dynamo) moving across a magnetic field. The current induced in the conductor reacts on the magnetic field so as to retard the motion of the conductor. This means that energy has to be expended in moving the current carrying conductor across the magnetic field. For a given velocity of movement the power required to move the conductor is directly proportional to the induced current. This explains how it is that more power is required to drive a dynamo if the current taken from it be increased.

The magnetic field diagrams given in Fig. 46 will serve to make this fundamental point clear. The magnetic field caused by the induced current (shown dotted in the left-hand diagram) reacts on the main field to produce the distorted magnetic field shown in the right-hand diagram. Clearly, the stretched magnetic lines are tending to force the conductor from right to left—that is, in the opposite direction to that in which motion actually occurs. In a dynamo, therefore, if there were no frictional or other losses, the power necessary to drive the machine would be determined entirely by the value of this electro-magnetic force acting on each conductor—that is, by the output of the dynamo in watts. The dynamo would then have an efficiency of 100 per cent.

7. **Eddy Current Brake.** Eddy currents, produced in electrical apparatus in which rapid flux variations occur, provide many problems for the electrical engineer to solve and, in certain types of apparatus, very special precautions are necessary to obviate these currents. On the other hand, eddy current effects are usefully employed for the purpose of *damping* or *braking*, particularly in certain types of electric meter. In Fig. 47 we show the essentials of an eddy current brake. When a metal disc is rotated between the poles of a magnet, currents are induced as shown, and we know that their direction is such as to oppose the motion producing them—hence the braking *action* on the disc. The actual effect of these eddy currents is to stretch the lines of force of the magnet in the direction of rotation, and as the magnetic lines resist this distortion, they exert a backward pull on the disc. The braking force is proportional to the speed, the thickness of the disc, and the square of the magnetic flux.

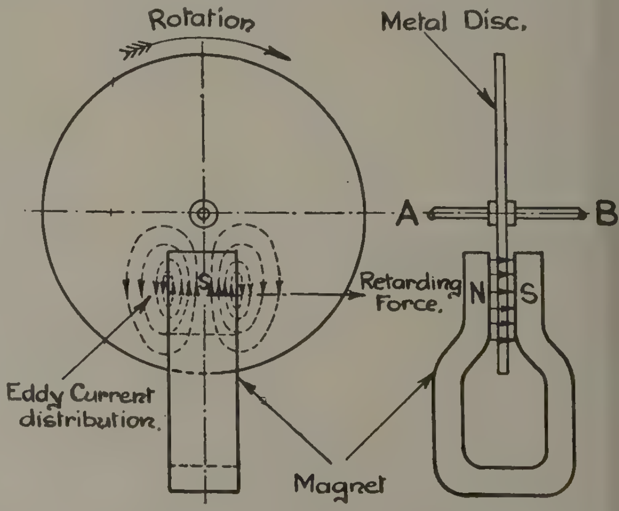

Fig. 47. Eddy Current Brake

RECIPROCATING STEAM-ENGINES

By Engineer Lieut.-Commander T. Allen, R.N. (s.r.)

LESSON VIII

STEAM CONSUMPTION AND THERMAL EFFICIENCY

Actual and Absolute Thermal Efficiency. Many important advances in the evolution of the steam-engine have had for their object the reduction of the steam consumption. The progressive nature of the improvement may be judged by the fact that Watt achieved a reduction of over 90 per cent of the consumption of early Newcomen engines; the triple expansion engine represented an improvement upon Watt to the extent of over 50 per cent, and further improvements have been made in recent years. Even to-day, however, a good condensing engine only converts into useful work about 20 per cent of the heat units supplied with the initial steam—the remaining heat being lost in the condenser cooling water (over 75 per cent) and by radiation.

The ratio $\frac{G}{R}$

$$= \frac{\text{rate of heat conversion to work in cylinder}}{\text{rate of heat supplied to cylinder}}$$

is known as the "thermal efficiency" of the engine.

The rate of conversion of heat energy to mechanical energy

$$G = \frac{\text{i.h.p.} \times 33{,}000 \times 60}{778}$$

$$= \text{i.h.p.} \times 2545 \text{ B. Th.U.'s per hour.}$$

Where 33,000 = work equivalent of 1 h.p. (ft.-lb. per minute).

778 = mechanical equivalent of heat (ft.-lb. per B.Th.U.).

The rate of heat supplied to the cylinder—

$R = Q \times S$ (B.Th.U. per hour) where.

Q = weight of steam supplied (lb. per hour).

$S = (E_1 - e_w)$.

E_1 = total heat per lb. of steam at pressure and temperature on boiler side of stop valve. (B.Th.U. above 32° F. From tables or charts).

e_w = water heat per lb. of water at temperature corresponding to pressure of steam in exhaust pipe (B.Th.U. above 32° F.).

The thermal efficiency η_{th} is therefore given by

$$\frac{G}{R} = \frac{\text{i.h.p.} \times 2545}{Q \times S}$$

It will be noted that this evaluation of the thermal efficiency only takes into account that proportion of the heat generated in the boiler which can properly be allocated to the cylinder, i.e. imperfections on the boiler and condenser sides of the cylinder are excluded. In considering the "absolute thermal efficiency" of the engine, the value of R takes into account the heat absorbed in generating steam at the pressure and temperature of the boiler, or super-heater, from water at the temperature of the boiler feed, or the air pump discharge. In these circumstances, the "absolute thermal efficiency" $\eta_a = \frac{G}{R_a} = \frac{\text{i.h.p.} \times 2545}{Q(E - e)}$

Where

E = total heat per lb. of steam at pressure and temperature of boiler, or superheater (B.Th.U. above 32° F. From charts or tables).

e = water heat of boiler feed or air pump discharge water above 32° F.

Ideal Thermal Efficiency. In the case referred to at the commencement of this lesson, $\eta_{th} = \cdot 2$, and at first sight this may appear to be an unfavourable result. It is shown in the lessons on the "Theory of Heat Engines," however, that the proportion of heat which can be converted into work, in an engine working on the ideal Carnot cycle between the absolute temperature limits of heat reception and rejection T_1 and T_2 respectively, cannot be greater than is represented by the ratio $\frac{T_1 - T_2}{T_1}$.

Standard Thermal Efficiency. It is, therefore, convenient to compare the performance of an actual engine with that of an ideal engine working on some standard cycle. For a steam-engine, the Institution of Civil Engineers recommends that the standard of comparison should be the "Rankine" cycle; defined by reception of steam up to cut-off at a pressure and temperature as measured on the boiler side of the stop valve, adiabatic expansion to the back pressure, and exhaust at constant pressure, as measured in the exhaust pipe close to the engine.

The thermal efficiency of this cycle $\eta_s = \frac{H_d}{S}$ is represented by the ratio between the work done H_d and the heat supplied S.

$H_d = E_1 - E_2$ where E_2 is the total heat of the steam after adiabatic expansion to the lower limit of pressure and temperature. The value of H_d may be calculated, or taken directly from tables of adiabatic heat drop. Vertical lines plotted on the Mollier diagram between the given pressure and temperature limits represent H_d to scale, and this application of the diagram is particularly convenient for practical calculations.

Efficiency Ratio. It will be noted that in calculating the thermal efficiency of an actual engine η_{th} and of the standard "Rankine" cycle of comparison η_s, the same limits of pressure and temperature are taken, and the ratio $\frac{\eta_{th}}{\eta_s}$ is named by the Institution the "efficiency ratio."

The efficiency ratio η_R is, therefore, expressed by the relation—

$$\eta_R = \frac{\eta_{th}}{\eta_s} = \frac{G}{R} \div \frac{H_d}{S} = \frac{\text{i.h.p.} \times 2545 \times S}{Q \times S \times H_d}$$

$$= \frac{2545}{H_d} \div \frac{Q}{\text{i.h.p.}} \quad . \quad . \quad . \quad (a)$$

The steam consumption of an actual engine C_a is usually expressed in lb. of steam per i.h.p. hour

$$= \frac{\text{total steam per hour}}{\text{i.h.p.}} = \frac{Q}{\text{i.h.p.}} \quad . \quad (b)$$

If expressed in lb. per b.h.p. hour, or lb. per kw. hour (for direct coupled generators) the steam consumption becomes—

$$\frac{Q}{\text{i.h.p.} \times \eta_{\textit{mech}}} \quad \text{and}$$

$$\frac{Q}{\text{i.h.p.} \times \eta_{\textit{mech}} \times \eta_{\textit{elect}} \times \cdot 746} \quad \text{respectively.}$$

where $\eta_{\textit{mech}}$ = mechanical efficiency of engine.
$\eta_{\textit{elect}}$ = efficiency of generator.

It should be noted that if the consumptions of engines working under different steam conditions are being compared, this notation is not an exact comparison of performance since the heat required to generate steam increases with the pressure and temperature. As already shown, however, the heat equivalent of work performed per h.p. per hour $= \frac{33{,}000 \times 60}{778}$ $= 2545$ B.Th.U.-hr., and the work done per lb. of steam supplied to an ideal engine working on the standard Rankine cycle $= H_d = E_1 - E_2$ B.Th.U. The steam consumption of this ideal engine C_i is, therefore, $\frac{2545}{H_d}$ lb. per i.h.p. hour . . . . . . (c)

Similar corrections as in the case of the actual consumption C_a may be made if the steam consumption is required in lb. per b.h.p.-hr., or lb. per kwh.

From expressions (c) and (b), the ratio between the ideal Rankine consumption and the actual consumption $\frac{C_i}{C_a} = \frac{2545}{H_d} \div \frac{Q}{\text{i.h.p.}}$ and from formula (a) this is equal to the efficiency ratio η_R. The efficiency ratio is, therefore, expressed by either of the ratios—

$$\frac{C_i}{C_a} = \frac{\text{ideal Rankine consumption}}{\text{actual measured consumption}} \text{ or}$$

$$\frac{\eta_{th}}{\eta_s} = \frac{\text{thermal efficiency (actual).}}{\text{thermal efficiency (Rankine)}}$$

Also, since $\eta_R = \frac{C_i}{C_a}$ it follows that C_a

$$= \frac{C_i}{\eta_R} \text{ or } \frac{2545}{H_d \times \eta_R} \quad . \quad . \quad . \quad (d)$$

Expressions (a), (c) and (d) are of considerable practical utility in predicting the probable consumption of a steam-engine. Many makers have derived values of efficiency ratio η_R from actual engine tests, and if the initial and final conditions are known, the value of H_d may be determined and the expression (d) evaluated to give the probable actual steam consumption C_a.

Thermodynamic Efficiency. This term is frequently used by engineers to define the ratio between the work actually obtained per lb. of steam, and the maximum work obtainable from 1 lb. of steam working on the standard cycle between the available limits of temperature.

By the notation employed in the lesson—

Work obtained per lb. of steam

$$= \frac{\text{i.h.p.} \times 2545}{Q} \text{ B.Th.U.}$$

Maximum work obtainable per lb. of steam
$= H_d$ B.Th.U.

∴ thermodynamic efficiency

$$= \frac{\text{i. h.p.} \times 2545}{Q \times H_d}$$

and from formula (*a*) this equals the efficiency ratio.

The value of η_r ranges from about ·45 for simple condensing engines using saturated steam to about ·85 for high-class simple engines supplied with superheated steam and exhausting against moderate back pressures. The value is greater for compound, triple expansion, and Unaflow condensing engines, than for simple condensing engines, and is likewise greater for non-condensing engines than for condensing engines.

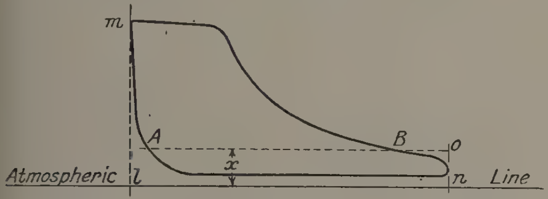

FIG. 25. INDICATED STEAM CONSUMPTION

Indicated Steam Consumption. The indicated consumption of an engine may be found from the indicator diagrams on the assumption that the steam is dry saturated. This computation does not, therefore, take account of the water in the cylinder, or of the loss by leakage, and if the indicated consumption is compared with the actual consumption, it will be found that the diagram only accounts for a proportion of the steam supplied to the cylinder. The difference between the indicated and actual consumption is usually referred to as the "missing quantity," and may vary from about 15 per cent of the total supply in high-class engines to about 75 per cent in small inefficient engines. A method of determining the indicated consumption is illustrated by the following example for a compound engine running at 70 r.p.m., stroke 5 ft., H.P. and L.P. net cylinder areas 738 and 3,000 sq. in. respectively. Indicator diagrams taken from both ends of the H.P. and L.P. cylinders show M.E.P.'s of 50·5, 49·5, 9·36, and 10·13 lb. per sq. in. respectively, giving a total power for the engine of 1,400 i.h.p.

Let Fig. 25 represent the indicator diagram for the front end of the H.P. cylinder.

Draw AB parallel to the atmospheric line, cutting the expansion curve at B prior to the opening of the exhaust valve to release; and the compression curve at A prior to the opening of the admission valve to lead. Draw a similar line on the diagram for the back end of the cylinder. Construct on both diagrams perpendicular bounding lines lm and no, so that ln represents the stroke or piston swept volume. Measure on each diagram the pressure represented by the height x of line AB above the atmospheric line. In the example—

$x = 25$ lb. $= 40$ lb. absolute for front end, and 27 lb. $= 42$ lb. absolute for back end. Mean $= 41$ lb.-sq. in. absolute. Measure on each diagram the ratio $\frac{AB}{ln}$ and find mean $= ·7825$. Then volume of steam accounted for by indicator diagram per revolution

$$= \frac{2 \times ·7825 \times 60 \text{ in. stroke} \times 738 \text{ sq. in.}}{1728}$$

$$= 40·1 \text{ cub. ft.}$$

Weight of 1 cub. ft. dry saturated steam at 41 lb.-sq. in. absolute pressure (from tables) = ·0996 lb.

Indicated steam consumption

$$= \frac{40·1 \times ·0996 \times 70 \text{ r.p.m.} \times 60}{1400 \text{ i.h.p.}}$$

$$= 11·95 \text{ lb./i.h.p./hour.}$$

A suitable addition should be made for the "missing quantity," when estimating the probable actual consumption.

Factors Affecting Steam Consumption. Improvement of the steam consumption is facilitated by—

1. *Running the engine at an economical ratio of expansion or cut-off.*

Theoretically, the greatest efficiency is obtained by adiabatic expansion from initial to back pressure, but the gain due to greater expansion increases the "missing quantity" by increased condensation and leak. Experience determines for each engine the most economical ratio of expansion, but considerable variation of the ratio is possible without marked change of consumption per i.h.p. hour.

2. *Compounding for normal type condensing engines.*

This is due to increase of expansion range without proportionate increase of "missing quantity." In compound engines the temperature and pressure range in each cylinder is reduced, with a corresponding reduction of initial condensation and leak, and only the leakage past the L.P. piston is lost to the condenser.

3. *Increase of initial pressure and superheating of initial steam.*

By increasing the work area per pound of steam and reducing initial condensation.

In modern practice, initial pressures between 160 lb. and 250 lb. are usual, but experimental work at much higher pressures is in progress, and it appears likely that the tendency of future reciprocating steam-engine design will be to utilize a higher range of pressure.

The increase in efficiency of actual engines using superheated steam is greater than the increase in the corresponding Rankine cycle of comparison. This is due to reduction of condensation and leak. The reduction in steam consumption for compound or Unaflow engines using steam at 150 lb. pressure superheated 200° F. is about 20 per cent of the consumption with saturated steam.

4. *Steam jacketing, reduction of leakage and radiation losses.*

External cylinder jackets with a free flow of steam are of value in small or slow speed engines in reducing condensation and leakage, but the reduction is greater for saturated than for superheated steam. Modern stationary engines are usually designed for superheated steam, and jackets are unnecessary on the H.P. cylinder. Where the valves and steam passages are situated in the cylinder covers, steam jacketing of the cylinder ends is achieved, and this is usual in good designs. Leakage may be reduced by attention to valves, packings, and piston rings, and radiation losses by efficient insulation of the heated surfaces.

Approximate normal steam consumptions for various types of engines are given in Table II. The consumption of larger engines is lower than that of smaller ones, and the influence of superheat is clearly illustrated by the table.

TABLE II

APPROXIMATE STEAM CONSUMPTIONS OF RECIPROCATING STEAM-ENGINES

Engine Type	Simple Expansion				Compound Expansion				Triple Expansion		Unaflow	
	Condensing		Non-Condensing		Condensing		Non-Condensing		Condensing		Condensing	
Initial steam pressure lb.-sq. in.	70–150		80–160		100–180		120–200		150–250		100–250	
I.H.P.	50–600		50–600		100–3000		100–3000		300–5000		150–3500	
Initial Steam Temperature	Sat.	600° F.	Sat.	600° F.	Sat.	600° F.	Sat.	600° F.	Sat.	600° F.	Sat.	600° F.
Steam Consumption lb./i.h.p./hr.	22–18	16–12	30–23	22–16	15–13	12–10	21–18	17–14	13–11	11½–9½	14–12	11½–9½

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON II

MEASUREMENT OF TEMPERATURE

Various methods of measuring temperatures are adopted in practice, each having its own particular field of application. The most important are—

1. Mercury thermometers.
2. Electrical resistance thermometers.
3. Thermo-electric pyrometers.
4. Radiation and optical pyrometers.

It is not proposed here to discuss the meaning of the term "temperature," or to show that the thermodynamic scale of temperature, first suggested by Lord Kelvin in 1848, is the basis of any true scale of temperature. The expansion of a perfect gas gives the same scale,[1] and therefore the International Bureau of Standards has adopted, as a standard thermometer of reference, hydrogen gas enclosed in a platinum-iridium bulb, to which standard all other thermometers may be compared.

Mercury Thermometers. These are so commonly used that it is hardly necessary to describe their construction, but there are certain points concerning their use in practice, and the errors to which they are liable, which call for some comment. Mercury thermometers are commonly used by engineers to measure the temperature of flowing fluids. It is necessary to place the thermometer well into the stream of fluid to avoid the influence of conductivity to or from the pipe. Preferably, the bulb should be immersed in the fluid itself by passing it through a gland in the pipe, unless the pressure of the fluid is high, but it is a common practice to screw a thermometer pocket into the pipe into which the thermometer is inserted. In this case the pocket should be deep and the walls thin, and to increase the thermal contact between the thermometer and the pocket a little mercury (or oil if the pocket is of brass) may be inserted into the pocket. This usually leaves a portion of the stem exposed to the atmosphere. If the thermometer has been calibrated with the stem fully immersed then the thermometer would read too low if the stem is exposed when the temperature to be measured is well above that of the atmosphere. This error may be allowed for by calculation[1] if thought necessary, but the correction for stem exposure may be avoided if the thermometer has been calibrated under the same conditions of stem exposure as when in use.[2] When the engineer is merely concerned with a comparatively small change of temperature, as in measuring the rise of the temperature of the circulating water in a surface condenser, the exposure of the stem is not of much consequence. When reading a thermometer care should be taken to avoid parallax, that is, the line of sight should be at the reading level and at right angles to the stem of the thermometer.

Special mercury thermometers for measuring up to 600° F. or 700° F. are constructed with an inert gas, such as nitrogen, introduced at a high pressure over the mercury before the stem is sealed off. This prevents distillation of the mercury into the upper part of the stem. But mercury thermometers for such temperatures are not very reliable, and if the stem is exposed to the atmosphere it should not be forgotten that the stem correction might be considerable, of the order 12° F. or 14° F. When a mercury thermometer is read an observation should be made of the stem to see that none of the mercury is detached in the stem. If found detached, usually the mercury may be induced to join the main column by jerking the thermometer violently downwards when held vertically in the hand.

Another form of the mercury thermometer is used in practice for measuring steam temperatures at boilers, superheaters, etc. The mercury is enclosed in a steel bulb inserted in the steam drum or pipe, and the bulb is connected to an indicator of pressure gauge type by means of a narrow bore tube. The pressure of the mercury increases with the temperature and moves the pointer on the indicator over the scale of temperature. Such an arrangement is represented in Fig. 1. Other fluids are also used in similar arrangements. The recording type of indicator is also obtainable where the pointer is arranged to give an inked record of the temperature on a circular chart driven by clockwork.

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[1] See Preston's *Theory of Heat*.

[2] *The Measurement of Steady and Fluctuating Temperatures*. Constable & Co.

Electrical Resistance Thermometers. It is well known that the electrical resistance of a metal wire usually increases with the temperature, though very few metals show a relation between resistance and temperature which can be relied upon when heated and cooled repeatedly. Siemens constructed a resistance thermometer formed by a platinum wire wound on a fireclay rod, but it was found to be unreliable as constructed. Messrs. Callendar & Griffiths found that this unreliability was due to contamination of the platinum under repeated heating, and that if pure platinum wire were kept free from all contamination it is one of the most reliable of materials for the measurement of temperature. These experimenters also determined very closely the law connecting the resistance of platinum wire with the temperature. The Cambridge Instrument Co., Ltd., is one of the best known makers of this type of instrument. The platinum wire is wound on a mica cross with leads of platinum, silver, or copper to the head of the thermometer, according to the maximum temperature for which it is to be used. In the Callendar arrangement compensating leads are also used of the same materials, and thus there are four terminals at the head of the thermometer. The platinum wire and leads are enclosed in a protecting sheath of silica, porcelain, or glass, depending upon the maximum temperature to be measured.

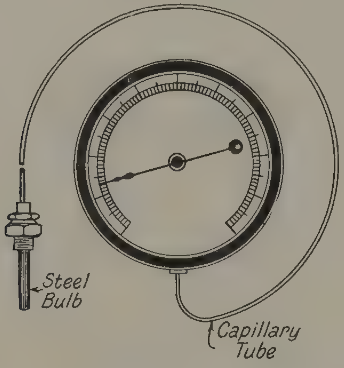

Fig. 1. Index Type of Thermometer

The measurement of the resistance of the platinum wire is obtained by a form of Wheatstone bridge, called the Whipple temperature indicator. The connections are represented diagrammatically in Fig. 2. The platinum coil T, with its leads P, are connected to one arm of the bridge, and the compensating leads C to the arm opposite. By this means the resistances of the leads are neutralized so far as they could affect the accuracy of the instrument, and actually all the four leads are plaited together between the thermometer head and the indicator. R_1, R_2, Z, BC, are standardized resistances. IB is a resistance, called the ice bobbin, and belongs to the particular thermometer being used. R is a small rheostat for adjusting the indicator to zero conditions. The reading is obtained in the following manner. When the battery switch or key BS is closed, a small current is supplied by the battery. If the potentials on the two sides, V and S, of the galvanometer G are equal, then the galvanometer needle remains in the neutral position, but if this balance does not exist the needle deflects to one side or the other. The slider S is then moved over the bridge wire BW and galvanometer wire GW until a point of balance is found. Thus, if the resistance of the coil T were to increase, due to a rise of temperature, the slider S would need to be moved towards the right in the diagram of Fig. 2. In the Whipple indicator the wire BW is wound spirally on a drum with the temperature scale also formed spirally on the drum. To secure a balance the drum is turned by a handle in the required direction, and the temperature is then read on the scale. The arrangement is capable of measuring from very low temperatures up to about 2,000° F. with great accuracy,

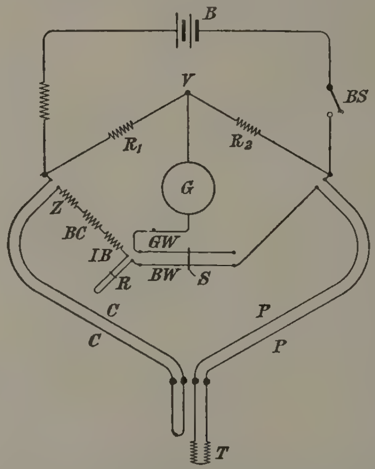

Fig. 2. Arrangement of Platinum Resistance Thermometer

but is rather costly to install, especially when the leads to the head of the thermometer are of platinum, as is required in measuring high temperatures.

Messrs. Siemens Bros. & Co., Ltd., build similar resistance thermometers, but use a three-wire method of compensation.

A cheaper arrangement, called the electrical distance thermometer, has been developed for measuring the temperatures of flue gases, steam, water, etc., at the various points required in a power station, or for measuring the temperatures of the refrigerator substance, and of the refrigerating rooms, in a refrigerator or cold storage plant, and for similar purposes in other industries. The thermometer coil is much smaller than in the previous arrangement, and is made to be inserted into pockets in much the same way as ordinary mercury thermometers, each thermometer being wired up to the indicator placed in a central position in the works. The arrangement, shown diagrammatically in Fig. 3, represents the practice of The Cambridge Instrument Co., Ltd. Accommodated in the same casing as the indicator *I* are the standard coils *a*, *b*, *c*, *d*, battery *B*, switch *S*, resistance *K*, switch *m*, and the balancing coils *n*. Each thermometer coil *t* is wired to its own balancing coil, and a common return connects all the coils *t* to the indicator. Each balancing coil is adjusted to give a standard resistance in the thermometer circuit, and thus allow for varying lengths of leads. The indicator shown in Fig. 4 is arranged for eighteen thermometers. The switch pointer in front shows which thermometer is in the circuit corresponding to the different positions of switch *m* in Fig. 3, and the heavy black line on the scale of temperature shows the pointer of the indicator. The switch is provided with an "off" and a "test" position.

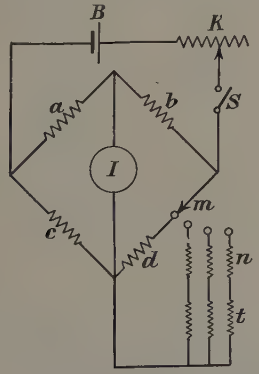

Fig. 3. Arrangement of Electrical Distance Thermometers

The indicator is operated in the following manner. Before taking readings of temperature it may be desirable to "test" the indicator reading. The switch is placed in the test position, that shown as *m* in Fig. 3, and if the indicator is correct the pointer comes to rest at a definite red mark on the scale, but if not correct the battery resistance, *K* in Fig. 3, is altered until the pointer comes to that definite position. On moving the switch *m* to the different thermometers the indicator pointer gives the respective temperatures by the positions read on the scale. In Fig. 4 an upper and a lower scale is provided, and a switch is operated to change from one scale to the other.

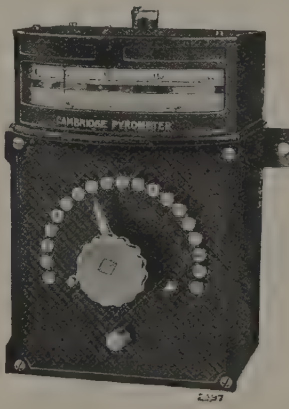

Fig. 4. Indicator for Electrical Distance Thermometers

Other makers of similar temperature indicators are Messrs. Siemens Bros. & Co., Ltd., Budenberg Gauge Co., Ltd., and the Foster Instrument Co. The last mentioned firm use a three-lead method of compensating for changes of lead resistance, and adopt a special coil control on the indicator to compensate for changes of battery voltage.

The thermometer coil or "bulb" may be constructed of nickel wire for temperatures up to about 700° F., but are preferably of platinum wire, especially for steam and flue gas temperatures.

Thermo-electric Pyrometers. When two wires of different metals are connected together, as at *H* and *C* in Fig. 5, and one end, say the end *H*, is hotter than the end *C*, a thermo-electric effect is obtained. The result is a small difference of potential which will drive a small current round the circuit, and the greater the difference of temperature between H and C, usually the greater is the flow of current. The difference of temperature between H and C is commonly shown by the deflection of the needle or pointer of a galvanometer over a scale of temperature. A common arrangement is represented in Fig. 6, where I is the indicator and C_1, C_2, are the terminals at the head of the pyrometer stem. If the leads C_1B_1, C_2B_2, are of the same materials as C_1H and C_2H respectively, then the cold junction is at the indicator terminals, but if ordinary leads of copper are used the cold junction is at C_1C_2.

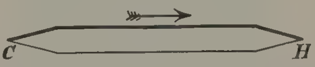

FIG. 5. THERMO-ELECTRIC COUPLE OR JUNCTION

If the temperature of the cold junction were kept constant then the indicator could be set to suit this temperature. In some indicators, connected as shown in Fig. 6, the indicator is specially constructed to allow automatically for any changes in the temperature of the cold junction. In other cases the cold junction C may be immersed in a thermos flask, or buried in the earth, or immersed in an electrically-controlled thermostat. Such an arrangement is represented diagrammatically in Fig. 7. The indicator is similar in appearance to that shown in Fig. 4.

The most reliable materials to form the junction is one wire of platinum and the other of an alloy of platinum and rhodium, and is suitable for temperatures up to 2,000° F. A cheaper material, the Hoskins couple, for similar temperatures is nickel with an alloy of nickel and chromium. Iron-constantan may be used up to about 1,300° F. For lower temperatures silver-constantan (about 930° F.) and copper-constantan (about 660° F.) are commonly used. These cheaper materials, sometimes called base metal couples, are suitable for measuring flue gas and superheated steam temperatures within the range of their application.

In most cases the resistance of the galvanometer or indicator has to be made large so as to swamp the possible variations of resistance of the leads. An arrangement sometimes adopted in laboratory work consists of a potentiometer, where the potential of the thermo-couple is balanced against a known drop of potential in a wire carrying a small current. The scale of the potentiometer can be tested against a known potential, such as that given by a cadmium or Weston cell. In this case the galvanometer may be of low resistance and the scale of the potentiometer gives the temperature when the galvanometer is in balance at the neutral position.

These various electrical thermometers or pyrometers are ordinarily subject to considerable lag when measuring variable temperatures and therefore are only suitable for temperatures which are nearly steady or which change slowly.

Radiation and Optical Pyrometers. These are the only instruments available for measuring furnace temperatures fairly continuously. In radiation pyrometers the total radiation emitted from a hole in the furnace is concentrated by a lens or parabolic mirror on to a small disc to which a thermo-couple is attached. The rise of temperature of the disc is a measure of the energy received, and the indicator connected to the thermo-couple gives the temperature of the furnace, having for this purpose been calibrated with the aid of the known laws of radiation. In the optical pyrometers the brightness or intensity of the light emitted is compared with the brightness of a metallic filament lamp in the instrument, and the means provided in the instrument for bringing the two brightnesses into unison is used to give the temperature. The instrument has to be calibrated against standard temperatures with the aid of the laws of light. It is not possible in the limited space available to go further into details or to discuss the methods used to calibrate the various thermometers and pyrometers which have been described. For further information and data reference may be made to the writer's work on *The Measurement of Steady and Fluctuating Temperatures.*

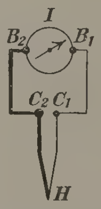

FIG. 6. ARRANGEMENT OF THERMO-ELECTRIC PYROMETER

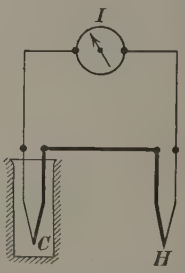

FIG. 7. ARRANGEMENT OF THERMO-ELECTRIC PYROMETER

DIESEL ENGINES

CONSTRUCTION, OPERATION, AND MAINTENANCE

By A. Orton, A.M.I.Mech.E.

LESSON VIII

MECHANICAL DETAILS OF THE FOUR-STROKE AIR INJECTION ENGINE

Pistons. A typical design of piston is given in Fig. 24, which shows a type suitable for cylinders up to about 16 in. diameter. Fig. 25 shows a similar type, but fitted with a separate top held to the main body by studs, designed to permit easy renewal in case of fracture. This figure shows the piston in its top position in the cylinder, at the moment of injection, and clearly illustrates the general shape of the combustion space, which may be roughly likened to a shallow segment of a sphere, with the injection valve nozzle placed at the centre of the flat side. The flats or slots cut in the top surface of the piston are to give clearance for the working of the air and exhaust valves. In Fig. 24 five rings are shown for keeping the piston pressure tight, but in practice the number varies from three to seven or even more.

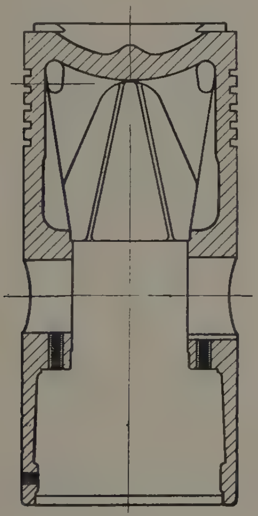

Fig. 24. Normal-type Piston

As indicated by the arrows in Fig. 25, heat is carried away from the piston to the water jacket by conduction through the cylinder walls. As the size of the piston increases, however, the flow of the heat is not great enough to keep the piston temperature down within permissible limits, and it becomes necessary either (1) to reduce the mean pressure in the cylinder, or (2) to adopt more positive means of cooling the piston. The first alternative involves a proportionate reduction in the

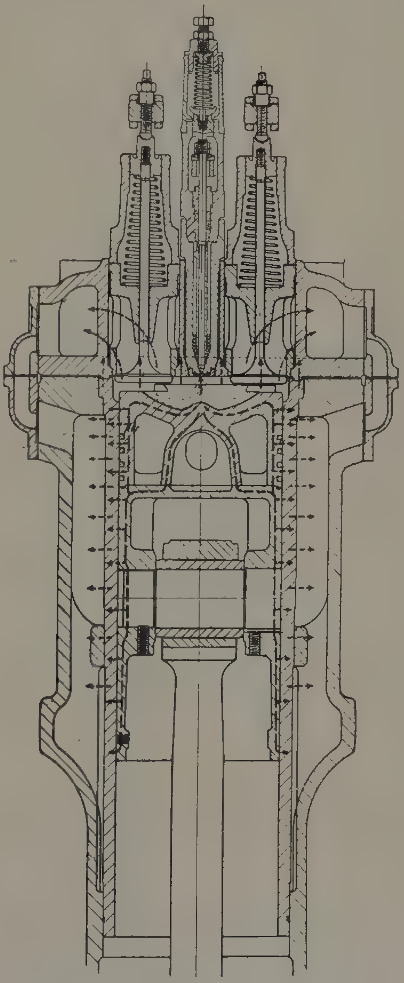

Fig. 25. Piston with Separate Top, Showing Heat Flow

power developed per unit of swept volume, but avoids the complication of special cooling devices. With larger pistons the second alternative cannot be avoided without great sacrifice in the power developed. By its adoption, however, the power rating of the engine can be maintained equal to that of the smaller normal engine. The exact size of piston at which a change in the system of cooling has to be made is, to some extent, a matter of choice, depending largely on the type and purpose of the engine. Broadly speaking, however, the change becomes necessary for pistons of over 20 in. diameter.

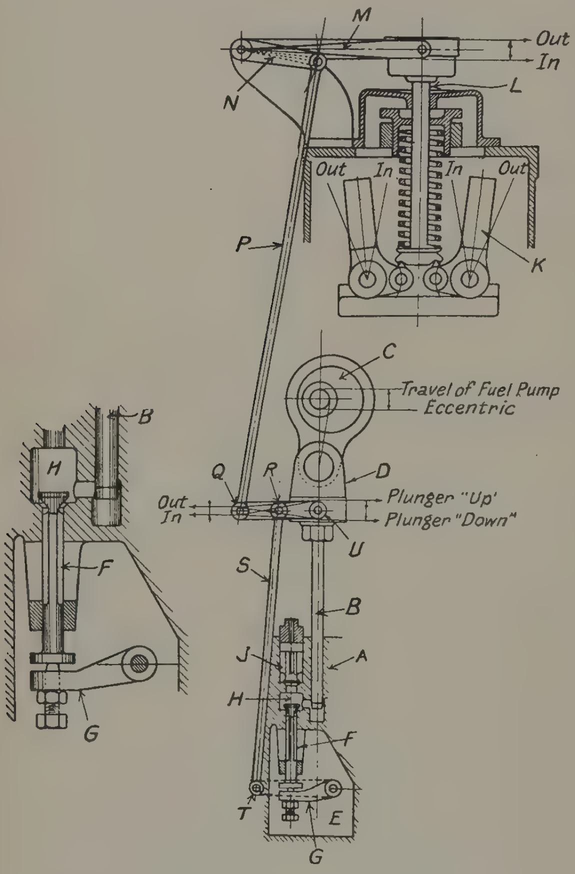

FIG. 26. DIAGRAMMATIC REPRESENTATION OF FUEL PUMP AND GOVERNOR

Positive cooling of the piston is obtained as follows: Immediately under the crown of the piston a closed chamber is formed, through which a cooling medium, either oil or water, is continuously circulated, thus carrying away heat and maintaining the material of the crown at a safe working temperature. There are many different arrangements of the mechanical parts for introducing the cooling medium into the piston, but space limitations here preclude a detailed description of them. At a later stage, illustrations will be given on which the cooling space between the head of the piston and also the connections for the cooling water or oil are shown.

It should be noted that both the piston and the piston rings are almost invariably made of cast iron of special quality, though aluminium has been used for the piston body to some extent on engines of the high speed type.

Fuel Pump and Governor. The function of the fuel pump and governor together is to deliver into the fuel valve casing, once per cycle, the correct quantity of oil for the particular load at which the engine is working, and to effect this with only a small increase in the speed of rotation from "full load" to "no load." In practice this increase can readily be restricted to within 4 per cent of the normal speed.

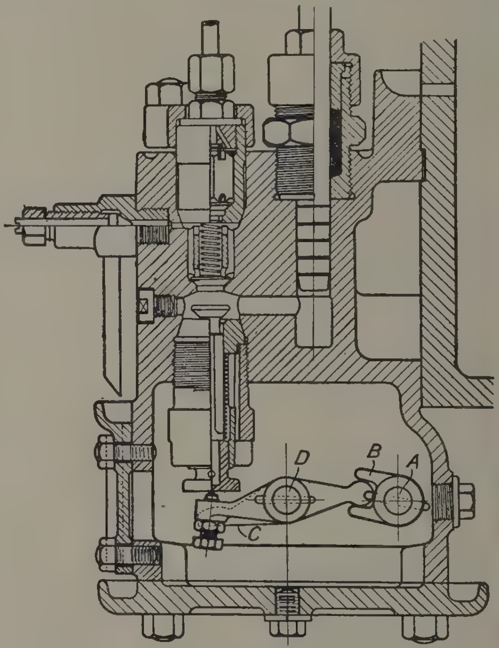

FIG. 27. DETAIL OF MIRRLEES FUEL PUMP

The essential features of the fuel pump and governor are illustrated in Fig. 26. In the fuel pump body *A*, the plunger *B* is reciprocated by the eccentric *C* and the suitably guided crosshead *D*. The eccentric is fixed on the camshaft and, therefore, gives the plunger one stroke per cycle. The lower chamber *E* is kept full of fuel oil by a gravity feed from the oil tank, and in it is situated the suction valve *F*, which is automatic as regards opening, but is controlled by the internal lever *G* as regards closing. On the upward stroke of the plunger *B* the suction valve lifts, admitting oil to the delivery chamber *H*, from which, on the downward stroke of the plunger, a certain portion of the oil displaced is forced through the two delivery valves *J* to the fuel valve in the cylinder head.

The centrifugal governor *K*, acting through the centre spindle *L*, levers *M* and *N* and rod *P*, controls the position vertically of the outer end *Q* of the horizontal lever attached to the crosshead. To the centre point *R* of this lever is attached the rod *S*, which is coupled at the lower end to an external lever *T*, fixed on the same spindle as the internal lever *G*.

When the engine is running steadily on load, the outer end *Q* of the crosshead lever is stationary, as the governor does not vary its position. The middle point *R*, therefore, moves vertically with the crosshead, but with half its stroke. It follows, therefore, that the end of the internal lever *G*, which engages with the lower end of the suction valve *F*, moves up and down exactly as the plunger moves up and down, but with a shorter stroke. The tappet screw in the end of the lever *G* is so adjusted that the suction valve is prevented from coming down on to its seat until the plunger has completed a certain portion of its stroke. During that portion of the stroke oil flows back through the suction valve, but after the valve has shut oil is forced through the delivery valves. Obviously, by varying the point at which the suction valve closes, varying amounts of oil will be pumped to the fuel valve, the earlier it closes the greater the amount, the later it closes the less the amount. The governor actually performs this function in the following manner.

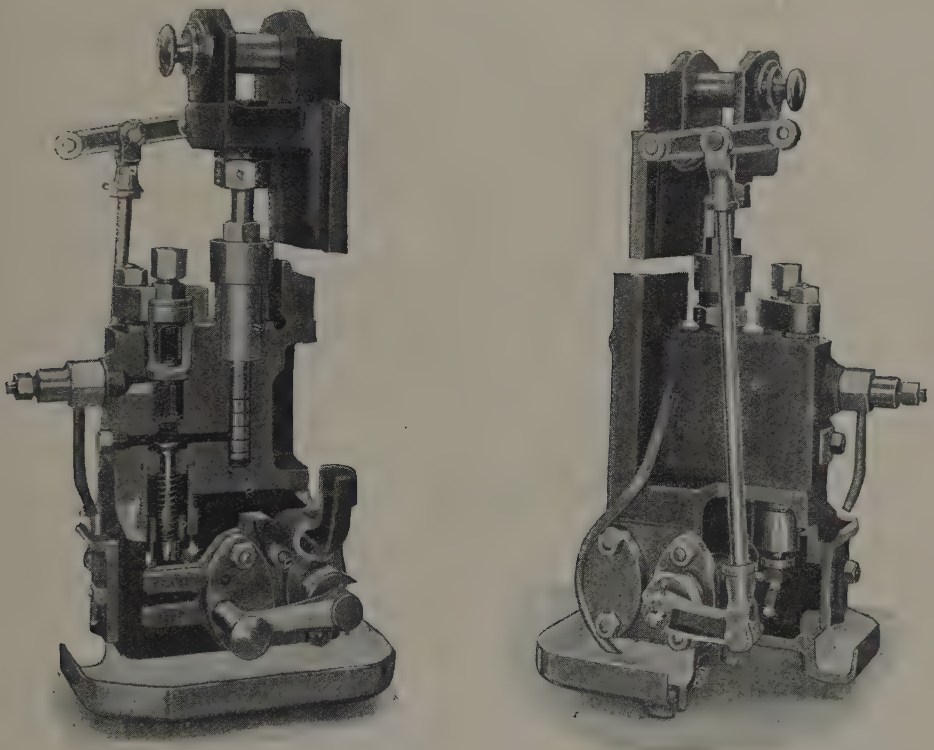

Fig. 28. Mirrlees Fuel Pump in Section

If the load on the engine decreases there is an excess of power developed causing the engine speed, and consequently the governor speed, to increase. The governor weights move outwards thereby raising the central spindle L, and with it the outer end Q of the crosshead lever.

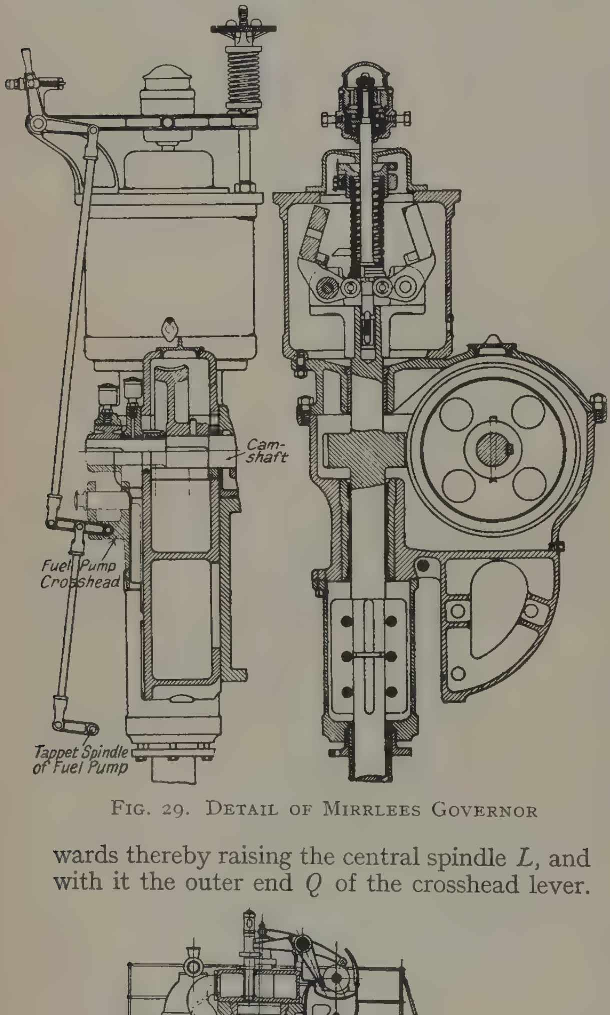

FIG. 29. DETAIL OF MIRRLEES GOVERNOR

This raises also the middle point R at all periods of its stroke, i.e. the top and bottom points of the stroke are both higher. The effect of this is to delay the moment at which, on its downward movement, the suction valve will reach its seat, and consequently to decrease the effective portion of the plunger stroke. Less oil is therefore delivered, the power decreases, and equilibrium is again established between the power developed and the load, the engine running steadily at a slightly increased speed.

If the load on the engine increases the reverse of the above process takes place, more oil is delivered, and the engine runs at a slightly lower speed. The relation between the fuel pump and the governor is such, that if the governor reaches its outermost position no fuel whatever can be delivered to the fuel valve. On the other hand, if the governor is at its innermost position, as when the speed is below the governing range, as it is during starting, the suction valve closes comparatively early in the delivery stroke and an excess of oil is delivered, causing the development of high power in the cylinder and a rapid increase in engine speed. It is to be noticed that the governor never has to *lift* the suction valve against pressure, as no pressure exists in the delivery chamber until the valve is down on its seat and the tappet lever below is not in contact with it. There is, therefore, no reaction from the fuel pump on to the governor, and this greatly facilitates smooth and perfect speed regulation.

The above principles of governing are adopted on the majority of Diesel engines by different

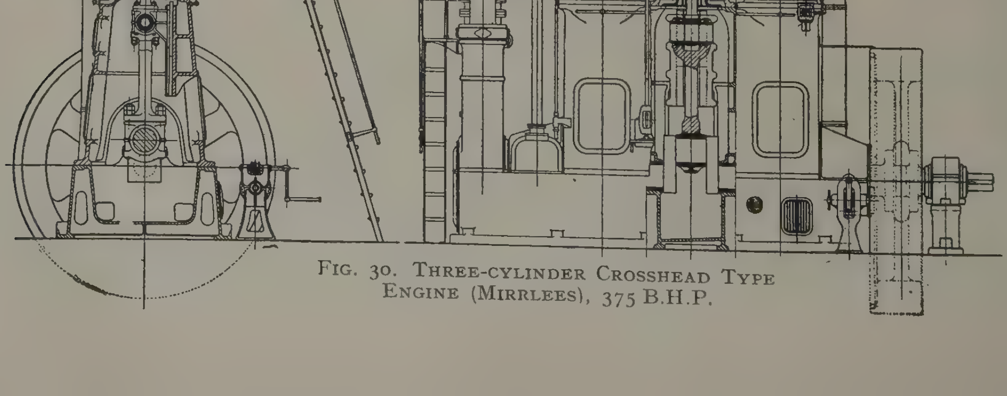

FIG. 30. THREE-CYLINDER CROSSHEAD TYPE ENGINE (MIRRLEES), 375 B.H.P.

makers though, of course, the mechanical details vary considerably. We give in Figs. 27, 28, and 29, sectional views of the fuel pump and governor as fitted on one of the Mirrlees engines, from which, together with the explanation already given, the actual details will easily be understood.

In order to stop the engine all that is required is to raise the suction valve. This is effected by a small movement of an outside lever attached to the spindle *A* (Fig. 27), which, by means of the jaw lever *B*, presses down the circular end of the lever *C* (fulcrumed freely on the tappet spindle shaft *D*, and thus raises the other end under the suction valve.

Cylinder Frame and Other Principal Parts. These parts are of the same general construction as those of any other reciprocating engine. They are, of course, designed to suit the working conditions peculiar to the Diesel engine, such as the high maximum gas load on the piston, the comparatively large inertia forces, due to the reciprocating parts and the heavy loads on the bearings. These, however, are only questions of correct mechanical design. The general construction of these parts is indicated clearly in the sectional drawings already given, and in Figs. 30 and 31 now given, showing engines of the crosshead type. Little further comment is therefore necessary. Attention may, however, be directed to the cylinder liner and its method of construction. Except in very small engines this most important part is always made separately as a plain circular casting, and is fitted into the cylinder casing or frame in such a manner that the lower end is free to expand axially, the upper end only being held firmly. The joint between the liner and the frame at the lower end is made by means of rubber rings fitting into grooves, or by some other form of packing which does not restrict any lengthwise expansion of the liner due to the high temperature at which it has to work.

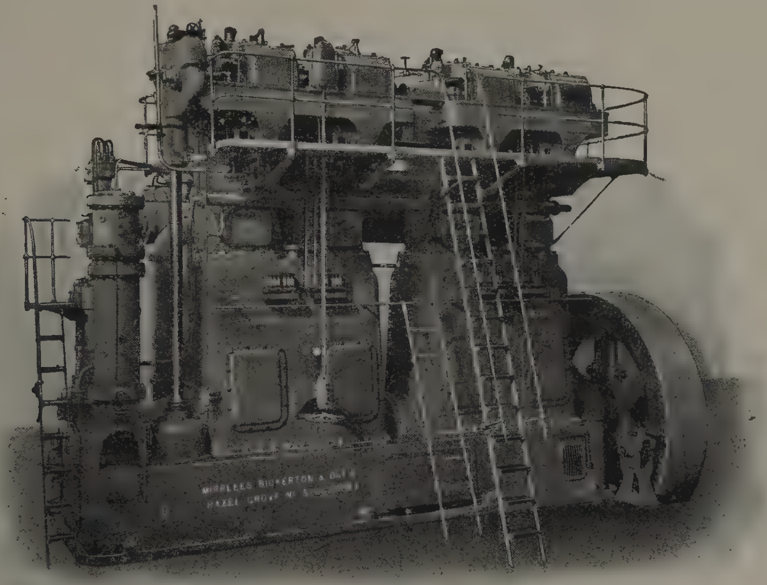

FIG. 31. 600 B.H.P. FOUR-CYLINDER "MIRRLEES-DIESEL" HEAVY OIL ENGINE. (CROSSHEAD TYPE)

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON I

INTRODUCTION

Whilst it would be presumption to suggest that it is possible to indicate in a treatise of this character all the many conditions which, more or less, control the elements of locomotive design, it is nevertheless a practical proposition to make an attempt to outline a few of the prominent features which must necessarily be considered in any design, or contributions to a design, of modern steam locomotives. It is essential, in the first instance, to ascertain exactly the duty required of an individual locomotive, the nature of the service in which it is to be engaged, the physical characteristics of the railway on which it is to operate, the tonnage it is expected to haul, and, generally, the conditions under which it is to be employed, with the available fuel and water supplies. In the following notes the leading features will be dealt with in sections, and the first of these will be devoted to consideration of the resistances a locomotive has to encounter due to grades, curves, and the condition of the track, with frictional resistance of the rolling stock, and opposition to traction imposed by atmospheric conditions. Adhesion, tractive effort, etc., will also be considered.

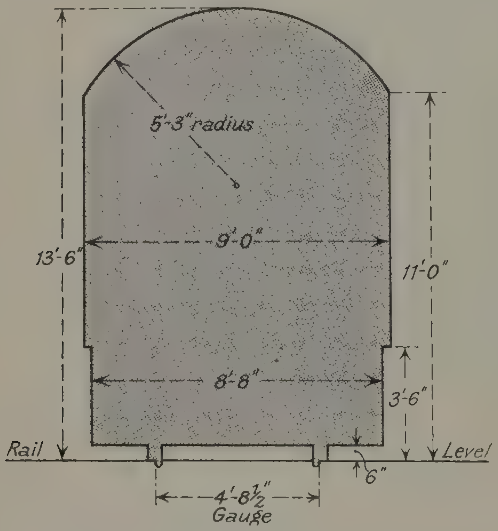

Fig. 1. British Loading Gauge

Resistance Due to Gauge, Grades, etc.

The gauge, or distance, between the rails on the track on which a locomotive is intended to operate must necessarily considerably influence its design, as also will the over-all dimensions allowed in construction and the permissible weights per axle, or per foot run.

In practice, narrow gauges tend to limit speeds, but not the traction or haulage power. Larger and more powerful locomotives are now running on railways of 3 ft. 6 in. gauge than are to be seen on the normal British lines, and, correspondingly, much heavier service is being performed on the 4 ft. 8½ in. railways of the U.S.A. than those of the 5 ft. 6 in. gauge in India; speeds, on the other hand, with the latter are distinctly medium.

There is the rolling friction of the train on the track, sliding friction (including the journals of the axles in their bearings), and the atmospheric resistance to the movement of the train—all calling for power to overcome them. Gauge seriously affects resistance problems, especially on curves. It also has its influence on the length of any rigid wheel-base that can be adopted. The wider gauges must necessarily require more tractive effort, and, conversely, as this decreases with the width of gauge, the minimum will be reached on a mono-rail.

Flexible wheel-bases, bogie trucks, central attachments in the shape of combined couplers, all conduce to the reduction of rolling and flange friction, or resistance, on curves.

The following deductions apply to railways of normal, 4 ft. 8½ in. gauge—

The resistance measured in pounds per ton weight to be overcome on a level railway by a locomotive varies with the speed—gradually increasing. For an ordinary express passenger engine it may be taken at, approximately, 10 lb. per ton weight at 10 miles per hour, rising to 30 lb. per ton at 60 miles per hour; whilst that for the train of passenger carriages behind it can be estimated at from 4·5 lb. to 14 lb. per ton of weight at the same respective speeds.

American observers quote 4 lb. per ton of train (engine and tender included) at 15 miles per hour; 4·8 lb. at 20, and 19·5 lb at 60 miles per hour.

The influence of grade will incur additions to these figures somewhat as follows—

On an incline of ·2 per cent, or rising 1 ft. in 500 ft., 4·5 lb. per ton; and on an incline of 1 in 20, approximately, 112 lb. per ton.

Additional resistance will be experienced due to curves, but it is customary to allow the reduction of speed automatically incurred to counterbalance this on railways having normal alignments. Special routes with continual curvature of undue severity must be specially dealt with by providing locomotives of increased adhesion and tractive power.

As an example of the estimated resistance encountered in hauling a passenger train of 10 coaches, weighing 340 tons, hauled by a locomotive weighing, with its tender, 100 tons, accepted data gives—

On a level line at a speed of 60 miles per hour.	340 tons weight of train × by resistance in lb. per ton at 60 m.p.h. = 13·4. lb,

which, added to the weight of engine and tender = 100 tons × the resistance in pounds per ton at 60 m.p.h. = 32·5.

$$340 \times 13{\cdot}4 = 4{,}556$$
$$100 \times 32{\cdot}5 = 3{,}250 \quad = 7{,}806 \text{ lb.}$$

On an incline of 1 in 100, at a speed of 30 miles per hour, this would be increased by some 80 per cent.

In the case of a goods train of 1,200 tons, hauled by a locomotive weighing 120 tons, at a speed of 30 miles per hour, on a level line, the resistance would be, approximately—

$$1{,}200 \times 8{\cdot}2 = 9{,}840 \text{ lb.}$$
$$120 \times 31 = 3{,}720 \text{ lb.} = 13{,}560 \text{ lb.}$$

(The figures "8·2" and "31" represent the accepted pull in pounds per ton for the type of locomotive and class of vehicle at the given speed.)

In similar manner in the case of the passenger train, the 13,560 lb. pull will become nearly 24,000 lb., when the train is ascending an incline of 1 in 100, and the speed drops to 10 miles per hour.

The above gives very elementary estimates of the resistances to be overcome by a locomotive in hauling trains under favourable conditions and at uniform speed. Further to these figures, there is the power required to gradually overcome the inertia offered by the engine and train in resistance to movement; in other words, the force required for acceleration.

From accepted data, it is found that a train weighing 450 tons (engine and tender included), starting from "rest" and reaching a speed of 50 miles an hour in a period of 6 minutes, encounters extra resistance as follows—

From 0 to 16 miles per hour in 1 minute – 12,960 lb. in a distance of 704 ft.
From 16 to 25 miles per hour in 1 minute – 7,290 lb. in a distance of 1,804 ft.
From 25 to 42 miles per hour in 2 minutes – 6,885 lb. in a distance of 5,896 ft.
From 42 to 50 miles per hour in 2 minutes – 3,240 lb. in a distance of 8,096 ft.

It will be seen from the above that the resistance to acceleration gradually diminishes as the speed increases.

It is generally accepted that an incline of 1 in 40 is practically a limit on main line railways, but 1 in 12½ is permissible for specially constructed lines relying on adhesion locomotives for traction; beyond this "rack" or other special devices must be adopted. The manner in which "UP" grades of a railway are compensated for by corresponding "DOWN" inclines will materially affect the steaming capacity of the boilers and power output of the locomotives employed.

Recent experiments have shown that large capacity vehicles for transport of merchandise offer much less resistance to haulage than small ones, and the more concentrated the load can be arranged behind the tender the better for the locomotive. Light wagons and carriages offer proportionately more resistance to movement than heavy ones.

As regards resistance due to climatic conditions, many authorities could be quoted on these, but it will suffice to say, all agree that the resistance due to side winds is much more than that imposed by head-end resistance. It would appear that there is much in this dictum, as examples of attempts to reduce the resistance of trains by adopting a "prow" like front to the locomotives have met with little or no success. Far more might be done, and, probably, is obtained, by close-coupling the trains, thereby reducing the "pockets," or air spaces, between the vehicles and the side superficial area generally.

To secure necessary adhesion, a locomotive must have weight, and the permissible amount of this is controlled by the strength of the permanent way, bridges, etc. It must be satisfactorily distributed on the driving wheels, otherwise it is possible for the power applied in the cylinders to cause the wheels to revolve, or "slip," without hauling the train. The resistance of vehicles to traction, as already mentioned, varies considerably with prevailing conditions, speed, and other factors; it may, however, be taken as from 4 lb. per ton on railways with good tracks, oil lubricated journals, etc., to 8 lb. per ton for traction under more indifferent conditions.

It is in the careful adjustment of the adhesive weight, power of cylinders, etc., where skill is required in design. The tractive effort of a locomotive may be defined as the "amount of pull" the engine can exert on the drawbar under good conditions of boiler pressure, correct expansion of steam in the cylinders, and a normal surface of the rails. The designer must bear in mind that he may have limitations imposed in the weight possible to be carried on each pair of driving wheels, and this renders it necessary to consider the number of drivers to be employed. Further, it is useless to provide more cylinder power than the weight of the engine will permit to be utilized as tractive power. It may be assumed that about one-fourth of the total weight of a locomotive on its drivers is available for adhesion, that is to say, the proportion of the load utilized to give a grip on the rails. Presuming, say, an 8-coupled engine to weigh 60 tons = 135,000 lb. approximately, then, on a dry track, some 33,750 lb. is available for adhesion, whilst if the same track is sanded, this will be increased to some 39,000 lb., but with wet or "greasy" rails, approximately, only 16,100 lb. will be available.

From the foregoing remarks it will be understood that there are many points which will require careful observation in any attempt to design a successful locomotive for specified service.

In the case of tank engines, allowance must be made in the available adhesive weight for the consumption of fuel and water, which causes a gradual reduction of load on the drivers. It is customary to calculate the adhesion of such engines on the net weight of the drivers with the bunkers and tanks empty. The disturbing influence of the amount of water in the tanks has also its effect on the steady riding and equilibrium of tank engines, and must be duly provided for.

A well-balanced distribution of weight on the axles of a locomotive is a proposition which requires much consideration and many calculations on the part of the designer. The method usually adopted for a new design is to ascertain the position of the centre of gravity from the estimated weights, and then endeavour to adjust matters to get as nearly as possible the desired distribution. Here a practical knowledge of working conditions is necessary for the designer.

As suggested, the balancing of a tank engine is more difficult than that of a tender engine, for the reasons given. The centre of gravity of such a locomotive intended for main line service may be taken as from 5 ft. to 5 ft. 6 in. above rail level.

Although tank engines were considered more liable to derailment, due to the disturbing influences, than tender engines, and were consequently seldom used for fast trains, such doubts have been removed by carefully recorded statistics, and now tank locomotives are often seen at the head of important express trains, running at high speeds, between centres comparatively near. They have the advantage of occupying less room in terminal stations, can run in either direction without turning, and can be arranged to possess unique powers of acceleration when required for rapid working of suburban passenger trains.

The centre of gravity of a locomotive having been obtained, the position of the axles to carry the load must be fixed, and herein the nature of the road bed, strength of rails and bridges, wil have the greatest effect, whilst the radius of the curves over which the locomotive will have to travel will determine the maximum rigid whee base permissible.

METALLURGY FOR ENGINEERS

By James Knowles

LESSON V

STEEL

The Cementation Process

A considerable proportion of the wrought iron manufactured by the Walloon and Swedish Lancashire hearth processes is dispatched to the Sheffield district, where it is treated as the raw material for the production of shear and crucible steels.

Despite the tremendous development of the Bessemer, Siemens open hearth and, latterly, the electric method of steel manufacture, the superiority of crucible steel for special purposes—notably high-class tools and cutlery—remains unchallenged. But, before this process can be adequately explained, it is first necessary to make a hurried survey, at least, of the production of converted or "cement" bar—once an operation of considerable importance.

The object of this process is to introduce carbon, to any extent, into pure Swedish iron.

It has already been explained that carbide of iron, when heated to a full red heat under oxidizing conditions, will give up its carbon. The cementation process is based on the converse of this, for iron, when heated in contact with carbon, with a total exclusion of air, will absorb it. Hence without entering into a description of furnace details, etc., it is sufficient for the present purpose to describe this method of carburization as follows—

Charcoal and Swedish iron bars are packed, in alternate layers, in firestone chests, which must be impervious to air, and sealed by an airtight arch of "wheelswarf." The temperature is then gradually raised to a bright red or yellow heat (1,000 to 1,100° C.) which is maintained for a period dependent on the degree of carburization desired. Carbon, being a migratory element, during cementation "flows" into the iron bars from the outside and, therefore, the depth of penetration and also the amount taken into solution obviously depends, in a large measure, on the time allowed. General examples may be quoted as: 8 days for mild (low carbon) heats, 9½ days for medium and, for hard heats, 11 days. Cooling down occupies any time from two to three *weeks*, after which the bars are unpacked.

In view of the foregoing, it will be readily understood that, considering the bars in cross-section, the percentage of combined carbon will be greatest at the outside and will diminish progressively towards the centre. A mild converted bar may actually present a core of unchanged iron ("sap") surrounded by steely layers which increase in hardness to the exterior. On the other hand, a harder heat may have a carbon percentage as high as 0·89 at the centre, when it is referred to as "steel through."

The effects of this penetration of carbon are most clearly shown by the following results of analyses of drillings taken at equally spaced points from the outside to the centre of the bars[1]—

(*a*) 0·95, 0·63, 0·39, 0·31, 0·15, 0·10 (centre) per cent carbon.
(*b*) 1·40, 1·26, 1·15, 1·10, 0·98, 0·88 (centre) per cent carbon.

It will at once be seen that the carbon contents of converted bars cannot be accurately determined by chemical analysis, and this gives the practical man an opportunity of beating the chemist. Converted bars, being brittle, are therefore broken and graded by the "bar sorter" who, with marvellous accuracy, bases his estimation of carbon percentage on the appearance of the fracture. These grades of hardness are referred to as "tempers" and, starting from the lowest, are numbered from 1 to 6, with occasionally ½ numbers marking intermediate figures.

No. 1 was usually known as "Irish temper," probably on account of its mildness, and the fact that no demand for it exists to-day may perhaps be held to constitute another glaring injustice. No. 6 bars are sometimes subjected to a second conversion when they are termed "double converted," or "glazed bars."

If, by any chance, air should gain admission to the iron during cementation, the carbon, at the high temperature prevailing, is burnt out on the surface of the bars. These are called

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[1] From notes of a lecture by Prof. J. O. Arnold D.Met., F.R.S.

" aired bars " and, being practically useless, are generally submitted to double conversion.

The approximate carbon contents corresponding to temper numbers differ considerably according to the practice of the works in which they are produced, but, as showing the incidence of numbers and tempers, the following may be taken as a good guide—

No. 2	0·60	per cent carbon
,, 3	0·80	,, ,,
,, 3½	0·85	,, ,,
,, 4	0·95	,, ,,
,, 4½	1·00	,, ,,
,, 5	1·35	,, ,,
,, 5½	1·45	,, ,,
,, 6	1·55	,, ,,

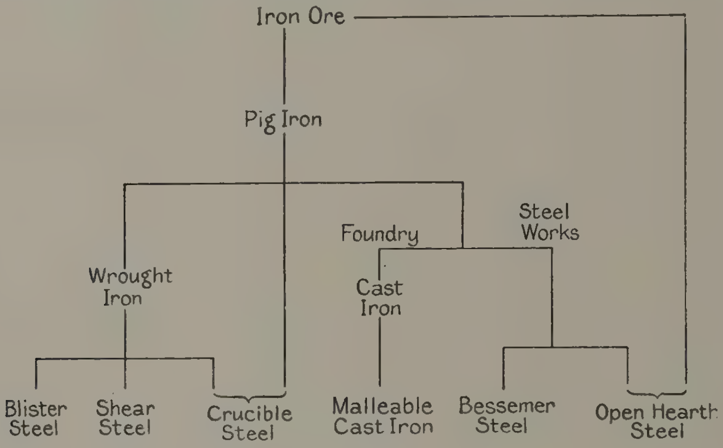


FIG. 5

Further reference must now be made to the slag inclusions in the Swedish iron. These consist essentially of ferrous silicate and, as the cementation operation proceeds, the penetrating carbon reacts with the oxides, reducing the iron to the metallic state and the silica to silicon, with a consequent evolution of carbon monoxide gas. The metal being in a plastic condition, this gas, in its efforts to escape, actually blows bubbles on the surface of the bars, producing blisters. The product of the cementation process is therefore known as " blister steel."

Shear steel owes its name to the insistence of the old-time cloth makers that their shears should be made of this quality of steel. Even at the present time it is still considered superior to any other for the manufacture of table cutlery and butchers' knives, etc., where a keen cutting edge should endure constant usage.

Blister bars, usually of No. 3½ or 4 temper, are raised to a yellow heat and hammered, thereby flattening out the blisters and imparting a certain amount of toughness to the otherwise brittle material. The " plated bars " thus made are cut into short lengths and piled into " faggots." These are covered with a suitable flux and clay to prevent decarburization, raised to a welding heat and drawn out under the hammer to specified dimensions. This is known as *single shear steel.*

Double shear steel is made from this by nicking the welded and hammered bars across the centre, doubling them over on to themselves and hammering out as before.

The weld lines always persist, notwithstanding the amount of work put upon the steel and, for this reason, shear steel is worked on the flat.

The course has now been cleared for a detailed consideration of the various methods of steel manufacture and, to help the student to understand the relationship between the various sections of the iron and steel industry, the accompanying chart (Fig. 5) has been drawn up. Accuracy of detail is not claimed on its behalf but, regarded in a broad sense, it is hoped that it will prove of some little assistance.

PATENTS FOR INVENTION

By W. E. Dommett, Wh.Ex., M.I.Mar.E., A.F.R.Ae.S.

LESSON VII

PATENTS OF ADDITION

When a patented device is being developed, it not infrequently happens that certain details call for modification, or additional methods of tackling the problem suggest themselves. Sometimes these developments are thought to be worthy of patent protection, and if of wide interest it may pay to take out a new patent. On the other hand, if the development is limited to the field covered by the original patent, then it is useful to remember that an application may be made for a "Patent of Addition." In this case the earlier patent is known as the "Parent Patent," and the new patent when granted runs with the earlier patent without payment of any extra renewal fees. The grant ceases when the parent patent expires. Since there are no renewal fees the total cost is £5. The modified Application Form 1c must be used, as there is an additional declaration that the invention is an improvement in or modification of the parent invention.

COGNATE APPLICATIONS

An inventor having filed a provisional specification may find his attention drawn along some other line of action in working out his ideas and, if not in a position to file a comprehensive complete specification, may deem it wise to file a second provisional specification, or even more than one extra provisional specification, for the new matter. It is allowable to file a comprehensive complete specification in respect of these provisional specifications, provided it can be drawn up to contain a single invention. A grant made on such a patent bears the date of the earlier provisional.

The two special forms of patent above referred to are not sufficiently numerous to justify detailed treatment, but, if the preceding lessons have been understood, the slight modifications in procedure called for in these special cases should present no difficulties.

MARKETING OF PATENTS

Assuming that an inventor is not in a position to manufacture and sell the articles which are protected by his patent grant, he has then to find some other means of obtaining a financial gain from the grant. Broadly stated, there are two methods open to him, one involving the assignment or transfer of the ownership of the grant, and the other consisting in the granting of licences whilst still retaining ownership.

Preliminary Steps. The circumstances of the case govern which of the two methods can be adopted to the greater advantage, but in either case it is necessary to get into touch with another party. This can be done by advertisement or direct approach. When advertising, the choice of the medium is important and, preferably, the trade paper concerned with the subject should be used. Usually, the advertisement can be backed by an article in the paper if the editor can be persuaded that there is sufficient of interest in the new idea to be of value to the readers. It is doubtful whether advertising is ever so successful as the method of direct approach. An introduction to a manufacturer, or man likely to take a financial interest, is far and away the most satisfactory first step, but failing an introduction or personal acquaintance with the desirable parties, a strong personality can often effect an entry into the desired circle with advantageous results. There are many striking, and in many instances amusing, examples of this method related about inventors who later achieved fame and wealth. One thing must be remembered most carefully, and that is, beware of the parasites who say they are wiling to help you place the invention, and who desire you to sign commission notes in return for services; if the proferred aid is such that on investigation it is worth having, then let the commission note be drawn up by a solicitor used to this type of business.

Licence of Right. Failing other means, it is open to the owner to apply to the Comptroller on Form 20 (Fee £1) for the grant to be endorsed under Sect. 24 of the Patents Acts, with the words "licence of right." When so endorsed an announcement is made in the *Illustrated Official Journal*, when all manufacturers know that it is open to them to manufacture, either on agreed terms with the inventor, or, failing agreement, on terms settled by the Comptroller.

Registration. Any action, either by way of assignment or licence or other form, should be notified to the Patent Office where it is registered, such registration being valuable in cases of possible later disputes and disagreements.

Assignments. When the rights are sold outright for cash, then the grant is transferred or assigned to the new owner. As to the equitable value of a sale outright for cash, no guiding lines can be laid down.

A favourite form of sale is one in which the inventor takes part of the value in cash and part in shares in a company to which the rights are transferred. There is one snag in this method in that the first cash payment may be the only one. Mismanagement or wilful wasting of assets or deliberate crabbing of efforts may result in failure, and the only guard against total failure is for the inventor to have a controlling number of shares or votes. Knowledge of this controlling power may prevent deliberate wrecking on the part of those anxious to get the full rights for a mere song on compulsory liquidation.

Syndicates. If sufficient capital cannot be raised for dealing with the manufacture and sale of a patented article, it is not unusual to form a syndicate which deals with the selling side, and the syndicate forms an alliance with a manufacturer willing to make the goods. This procedure can be extended to an alliance with a firm willing to handle the sales, in which case the syndicate merely holds the rights and gets its return from royalties. Really the syndicate then only needs the money requisite for getting into touch with the other parties—money which the inventor is not able to produce on his own. Also, this preliminary syndicate is valuable if the men brought into it have a better business capacity or more intimate knowledge of the particular trade than the inventor.

Royalties. The inventor, or a preliminary syndicate, may choose to get his returns by granting an exclusive licence to a manufacturer, or may grant general licences. In the latter case a flat royalty of from 5 to 10 per cent is agreed upon, and the three points to watch in this connection are: (1) that the price upon which the royalty is based is clearly determined. A specific statement that it is the list price, or the manufacturers' selling price, or one of the definite prices charged during the sale and resale from manufacturer to wholesaler, retailer, and final purchaser; (2) that the prices fixed are subject to the inventor's agreement, otherwise it may be fixed so high as to kill sales; (3) that special contracts are to be the subject of special consideration, it being advisable to agree to lower royalties on big contracts produced perhaps at a cut price to save probable loss of the contract.

Usually in licences, the inventor agrees to let the manufacturer have the benefit of any future improvements he may evolve.

Where an exclusive licence is granted, it is usual, in addition to the points put forth above, to include a clause in the agreement that during the first few years of the patent a scale of minimum royalties shall be paid. For example, suppose a new inner tube for bicycles and cars is being dealt with, and that it is considered the sales should be 1,000, 3,000, 10,000, valued at £500, £1,500, and £5,000 respectively during the first three years, then the royalties would be £50, £150, and £500 respectively on a 10 per cent basis, and such sums could be used in the agreement. Generally it is on this clause that the largest amount of haggling and time wasting occurs. It is advisable not to waste too much time, but to agree to reduced amounts, if necessary, providing one is confident of the merits of the invention and reasonably satisfied as to the *bona fides* and capacity of the licensee. One other clause relates to the right on the part of the inventor to withdraw the licence after giving due notice in cases where the manufacturer is not doing the best with the invention—an obvious case, perhaps, being that where the manufacturer has taken up a newer and better competitive article.

The special clauses mentioned are, of course, accompanied by a number of ordinary clauses common to all business agreements, which can be safely left to one's solicitor for their proper inclusion.

(CONCLUSION)

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XVI

ENGINEERING MATERIALS IMPROVED

BENJAMIN HUNTSMAN (1704–1776)

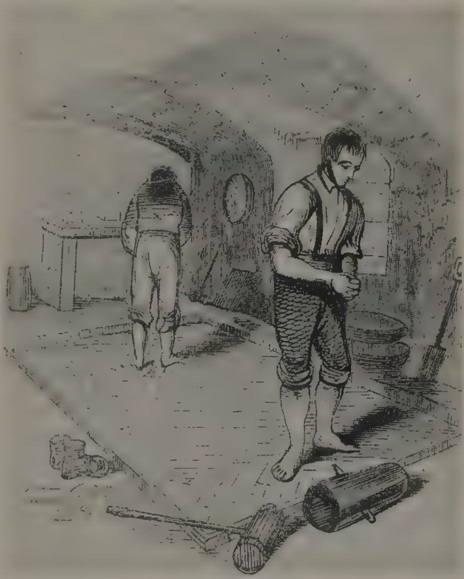
Curiously enough, the name of Benjamin Huntsman, the inventor of cast steel, is very little honoured at the present day. And yet Huntsman, by his one great discovery of a method for producing hard and consistently reliable steels, conferred an enormous benefit upon the engineering industry.

Early Method of Casting Steel Ingots
(From an Old Print)

One of the great practical difficulties which beset the path of all the early engineers was that of procuring reliable metal for the construction of their engines. Until the middle of the seventeenth century, iron and steel were prepared by methods which were, to say the least, centuries old, and, particularly in the case of steel, no bar of metal was ever manufactured in a condition which made it entirely homogeneous throughout its mass. Metals were soft during the early days of Watt and his co-workers, and, being soft, they soon wore out, and gave rise to trouble upon trouble.

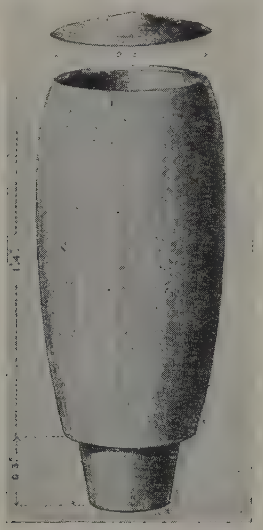
The evolution of the method of producing reliably hard grades of steel, which was due solely to Huntsman removed, once and for all, many of the above troubles, and it became possible for the practical engineer to manufacture more reliable and lasting types of machines.

Huntsman's method of manufacturing cast steel forms the basis of the present industrial process for the production of that commodity. In its essentials, the crude metal, in the form of "small round lumps," was placed in carefully annealed earthenware crucibles, or "pots." The "pots" were then packed into enclosed casting furnaces, the spaces between them being filled up with a mixture of coal and cinders. Here, the pots with their metal charges were allowed to remain for varying periods of time. Finally, they were withdrawn from the furnaces, and their molten contents run out into clay moulds.



Preparing Clay for the Making of Steel Moulds
(From an Old Print

The inventor of this process, Benjamin Huntsman, was born in Lincolnshire in 1704. He was a watchmaker by trade, and, during his early years, he augmented his income by practising medicine and surgery. It is said that his skill as a surgeon, and particularly as an oculist, was very considerable.



A Crucible, or "Pot," Used in Huntsman's Process for the Manufacture of Cast Steel

Huntsman was attracted to the manufacture of improved grades of steel by observing the consistently bad quality of that material which the Sheffield manufacturers turned out at that period. He devoted some years to experiment, and finally he commenced manufacturing his hardened cast steel at Handsworth, near Sheffield, about the year 1740.

The Sheffield steelmakers, however, would have little to do with Huntsman and his new steels, and it was only after the inventor had disposed of large quantities of his steel to French manufacturers, and the productions of those firms had entered into stringent competition with the Sheffield steel producers, that the latter began to display an active interest in Huntsman and his products.

Huntsman kept his process a secret. It is said, however, that the secret was exposed eventually owing to the tactics of two workmen who obtained an entry into Huntsman's works, disguised as casual labourers.

Death overtook Huntsman in 1776, but by that time his hardened steels had attained a considerable measure of renown and, after the death of the inventor, the steel manufacture was carried on, greatly extended and modified, by his son, William Huntsman.

WORKS ORGANIZATION AND MANAGEMENT

By E. W. Workman, B.Sc., A.M.I.E.E., etc.

LESSON VIII

THE PRODUCTION DEPARTMENT

Functions of the Production Department. At first sight the name of this department is apt to be a little misleading, as the main function of the department is not actual *production*, but the *control* of production from the point of view of time and urgency. The necessity of having some centralized control to ensure that all necessary parts are being manufactured so as to be ready when required for assembly, is very apparent in a large works. When it is remembered that, for example, in the case of a large turbine, upwards of 25,000 separate pieces are required to make one finished article, and that these pieces will range in size and quantity from a single 10-ton cylinder casting to three thousand or more steel blade rivets, it is seen that the problems involved in regulating the speed of production may be extremely complicated. The first function of the production department, therefore, resolves itself into the ensuring that delivery dates of customers' orders are not delayed by reason of late manufacture of any parts. This part of the problem is further complicated by the necessity of speedily replacing parts which prove defective during the course of manufacture.

The second function of the production department consists in the ensuring that the minimum amount of machinery is idle. This entails the proportioning of work so that facilities, such as cranes and transport, are not overloaded at any time, so as to cause an accumulation and consequent stoppage of the flow of work.

The combination of these two functions often calls for a very careful decision, as the rival demands will often point in opposite directions, and the correct solution of the problem may be by no means clear. The staff must, therefore, be experts in control and organization rather than have especially good technical qualifications.

Location of Production Department. As the production department is solely concerned in controlling the different processes of manufacture, it should be located in as central a position as possible, thus ensuring close contact with the continuously changing state of the factory production. In a large factory it will probably be found necessary to have a central production department with sub-offices in the chief manufacturing departments, as otherwise the production department would be too far removed from some of the shops to be able to work efficiently. At the same time the main production department should not be too far away from the other clerical departments from which it derives its information as to requirements.

Varieties of Work Orders. All orders for work must pass through the production department so that they may be issued to the shops in the right order of precedence having regard to all other work on order at the same time. The different kinds of order that may be received are as follows—

1. The general order, being the details of customers' and finished warehouse orders. This forms the most important type of order, and controls the entire external production of the factory.
2. The stock or stores order, for the manufacture of parts made in bulk for later transference to general orders.
3. The plant order, for the making of equipment to be used in the factory.
4. The repair order, for the upkeep of plant and equipment.

Material for Orders. In order that the department may be in a position to control the dates on which work is to be started on the different parts it is necessary that it shall be in close touch with the department controlling the supply of material, tools, and equipment. It must know what supplies are required and what supplies become available. It must decide the urgency and order of preference in obtaining outside supplies in the same way that it decides these questions inside the factory.

Information Required by Department. Before the production department can deal with any order it is necessary for certain definite information to be received. This information usually consists of the following documents—

1. The order form. This gives general particulars, such as the quantity of articles required and the dates on which delivery has been promised.
2. The drawings, both general and detail.
3. The specification sheets, giving a detailed list of all material and parts required.
4. Planning department information.

In the case of a large order, work may be commenced before the last three items have been completely received, as the detail work involved may be very great and require a corresponding length of time. In such a case, work would be started on those parts which required longest time to manufacture, and the details relating to the remaining parts would follow later, thereby considerably hastening the ultimate delivery date.

Routine Involved. When the above information has been received the production department can commence its organization of the details of manufacture. In the first place it is necessary to determine what parts actually need manufacture or buying outside, as quite a large number of items may be carried in the general stores and thus be available without special manufacture. That a part is *usually* carried in stock does not mean, of course, that the part will necessarily be available in sufficient quantity to supply any particular order without further bulk manufacture, and so the orders for stock must be linked up and controlled according to the demands made upon the stocks. That a part is *usually* carried in stock is very often indicated by the drawing department in the bill of material or specification sheet. Under a good system, orders for work to be done will not be issued until the material is ready or actually delivered to the place where the work is to be done. If necessary, this gives the production department the opportunity of substituting other material for any which has been delayed, and for altering the production programme for the other orders consequently affected.

Controls of Tools. In the same way that it is necessary for the material to be ready before orders to commence work are issued, it is equally necessary that it shall be ensured that all tools needed are available at the time when work is to be commenced. This applies especially to new tools which have been specified in particular cases by the planning department. For this reason the production of the tool-room must come under the control of the production department as regards deciding urgency and precedence. The orders for the making of tools will then be treated in much the same way as the orders for making the normal manufactures.

Issue of Instructions. There are several different ways of issuing instructions to the shops to commence work. The means adopted should give an automatic report when the piece has been completed, so that the production department is kept continuously in touch with the progress of the work. The commonest method adopted is to issue an instruction card for each operation or group of operations to be done. When the operations written on the card have been completed, the card is returned to the production department who thereupon issue the next card required for carrying on the manufacture of the article. This production card is very often the same as the one used for recording the employee's time, the one card performing two different functions in the production and cost departments. The method of issue of these cards, together with the specification sheets and material requisitions, allows of the exercise of considerable ingenuity in the layout of the forms required, and in clerical means of duplication. With well thought out organization an enormous amount of repetition clerical work may be saved—with a corresponding speeding up in the routine.

Filing of Instruction Cards. If the above system is in use, the file of unissued cards forms a very convenient reference index of work waiting to be done. As the number of cards is apt to be large in a big works, the careful and systematic filing of them is important. Those cards which represent work held up for any reason should be filed separately, so that they will receive special care and attention with a view to removing whatever difficulty is causing the obstruction.

It is convenient to group the cards representing work held up according to the class of obstruction, lack of tools, lack of castings, etc. The remaining cards, representing work which is ready for issue, may be divided conveniently into three classifications—urgent work, ordinary work, and work which should be delayed if other work is available. Under each classification the cards are arranged in order of urgency. A difficulty which arises, however, if the cards are filed in order of urgency is, that it is not easy to trace the cards belonging to any particular order number if required for alteration. The best method of filing the cards will vary with the different conditions found in different factories.

Charts. In addition to the above card records, it is usual to keep graphical records of the progress of the more important work. The latest date at which the work on each part should be commenced may also be found graphically by plotting each part against the number of weeks required for manufacture, commencing with the final assembly and working backwards, subtracting the number of manufacturing weeks for each part from the date when that part is required in the assembly or part assembly of a portion.

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MATHEMATICS FOR ENGINEERS

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LESSON XVIII

TRIANGLES

The triangle is the most important geometrical figure having straight sides, and any other rectilinear figure is usually considered, when calculations, etc., have to be made about it, by division into a number of triangles. In the preceding lessons we have been able to calculate unknown sides and angles of triangles from sufficient data, by dividing, as is always possible, any triangle into two right-angled ones. On account of the many occasions when we have to perform such calculations, it will be a great saving of time and labour if we find, for use upon these occasions, the relations which exist between the sides and the trigonometrical ratios of the angles for any triangle. There are two extremely important fundamental relations, one involving the sines of the angles, and the other the cosines.

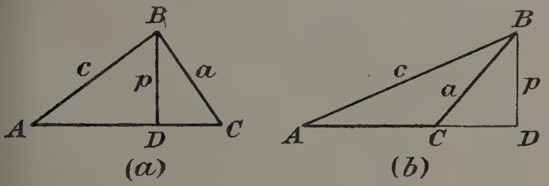

Fig. 76

Sine Rule. Let ABC be a triangle (Fig. 76), and draw BD perpendicular to AC. So that our proof may do for *any* triangle, we have drawn two cases, one, (*a*), in which $\angle C$ is acute, and the other, (*b*), in which $\angle C$ is obtuse. Calling BD p, and denoting, as usual, the lengths of the sides by a, b, and c, we have

(*a*)	(*b*)
From triangle ABD, $p = c \sin A$	$p = c \sin A$
From triangle CBD, $p = a \sin C$	$p = a \sin \angle BCD$ $= a \sin (180° - C)$ $= a \sin C.$

Thus, in *both* figures, equating these values of p, we have,

$$a \sin C = c \sin A$$

or,
$$\frac{a}{\sin A} = \frac{c}{\sin C}$$

By drawing the other perpendiculars, we can show in exactly the same way that each of these is equal to $\frac{b}{\sin B}$, so that, for all triangles.

$$\frac{\mathbf{a}}{\sin \mathbf{A}} = \frac{\mathbf{b}}{\sin \mathbf{B}} = \frac{\mathbf{c}}{\sin \mathbf{C}} \qquad . \qquad . \quad (A)$$

There is another way of obtaining this, which also leads to a new fact of some importance. Consider a triangle ABC inscribed in a circle (Fig. 77). Let BD be the diameter through B, and join DC. Then we have $\angle D = \angle A$, since they both stand on the arc BC; also, $\angle DCB = 90°$, since it is the angle in the semicircle BCD. Denoting the radius of the circle by R, we have $BD = 2R$, and then, from the triangle BCD,

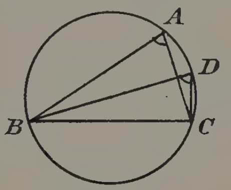

Fig. 77

$$BC = BD \sin \angle BDC$$

or,
$$a = 2R \sin A$$

whence
$$\frac{a}{\sin A} = 2R.$$

Consequently,

$$\frac{\mathbf{a}}{\sin \mathbf{A}} = \frac{\mathbf{b}}{\sin \mathbf{B}} = \frac{}{\sin \mathbf{C}} = 2\mathbf{R} \qquad . \quad (A')$$

We may state the Sine Rule (A) in words, that *the sides of any triangle are proportional to the sines of the opposite angles.* This is by far the most important and most frequently used of the formulae for the sides and angles of a triangle, and whenever we know one side of a triangle and the angle opposite it, this rule will enable us to "solve the triangle," i.e. with the help of one other element (of course, three in all are necessary), to calculate the remaining sides and angles. The fact that each of the fractions in (A') is equal to $2R$ is sometimes useful, and the relation $a = 2R \sin A$ can be used for any chord of a circle—*the length of any chord of a circle is found by multiplying the diameter by the sine of the angle subtended at the circumference.*

EXAMPLES. 1. In a survey, a point C is observed from two others, A and B, 352 yd. apart. The angles BAC and ABC are found to be 36° 41′ and 44° 33′ respectively. Find the distances of C from A and B.

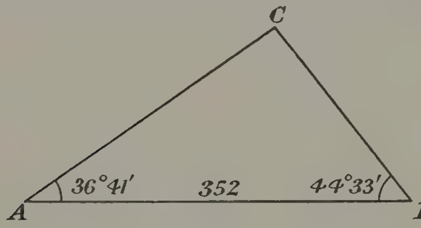

FIG. 78

The data are shown in Fig. 78. We first find the angle C, which is

$$180° - (36° \, 41' + 44° \, 33') = 98° \, 46'.$$

By the sine rule,

$$\frac{a}{\sin 36° \, 41'} = \frac{b}{\sin 44° \, 33'} = \frac{352}{\sin 98° \, 46'}$$

As the angle C is in this case obtuse, we must use the sine of its supplement, 81° 14′, and so we have,

$$a = \frac{352 \sin 36° \, 41'}{\sin 81° \, 14'}$$

$$= \underline{212\cdot8 \text{ yd.}}$$

2·5465 (*log.* 352)
$\bar{1}$·9949 (*log. sin* 81° 14′)
2·5516 (subtracting)
$\bar{1}$·7763 (*log. sin* 36° 41′)
2·3279 (adding, = *log. a*)

$$b = \frac{352 \sin 44° \, 33'}{\sin 81° \, 14'}$$

$$= \underline{249\cdot9 \text{ yd.}}$$

2·5516 (from above)
$\bar{1}$·8461 (*log. sin* 44° 33′)
2·3977 (adding, = *log. b*)

With regard to the logarithmic calculations, we should notice that the denominator in both of the fractions is the same (*sin* 81° 14′), and also that there is 352 in each numerator. Consequently, it will be quicker to do this division *once only* and to do it first. Again, most sets of tables contain logarithmic sines, that is, the logarithms of sines, and so save the trouble of looking up first the sine and then its logarithm. The rest of the work should be quite clear.

2. The crank of an engine is 9 in. long, and the connecting rod 40 in. long. Find the distance moved through by the piston as the crank turns 65° from the inner dead centre.

In Fig. 79, AB is the crank, BC the connecting rod. D is the end of the stroke, where $AD = 40 + 9 = 49$ in. We use the triangle ABC to find AC, and then obtain the required length, DC, by subtraction.

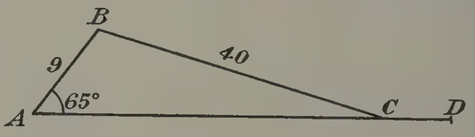

FIG. 79

The sine rule gives

$$\frac{40}{\sin 65°} = \frac{b}{\sin B} = \frac{9}{\sin C}$$

From the first and third we have

$$\sin C = \frac{9 \sin 65°}{40}$$

whence

$$C = 11° \, 46'$$

so that

$$B = 180° - (65° + 11° \, 46') = 180° - 76° \, 46';$$

therefore,

$$\sin B = \sin 76° \, 46'$$

·9542 (*log.* 9)
$\bar{1}$·9573 (*log. sin* 65°)
$\bar{1}$·9115 (adding)
1·6021 (*log.* 40)
$\bar{1}$·3094 (subtracting = *log. sin C*)

Using this in the first and second fractions in the sine rule gives

$$b = \frac{40 \sin 76° \, 46'}{\sin 65°}$$

$$= 42\cdot96 \text{ in.}$$

Then,

$$CD = 49 - b = 49 - 42\cdot96$$

$$= \underline{6\cdot04 \text{ in.}}$$

1·6021 (*log.* 40)
$\bar{1}$·9883 (*log. sin* 76° 46′)
1·5904 (adding)
$\bar{1}$·9573 (*log. sin* 65°)
1·6331 (subtracting = *log. b*)

The only point to note in the above calculations is that we have not looked up the antilog. in the first part, but found the value of C direct from tables of log. sines.

There is, however, a very important point of principle that the student may have noticed. In the first part, we have found an angle from its sine, and in such a case there are always *two* possible angles less than 180°, the one actually found in the tables, and also its supplement. Our knowledge of the practical features of the problem makes it quite certain that the *acute* value of C is needed, so we have not bothered with the other, as being impossible in the practical problem. This raises two points: (1) in some instances the double answer *will* be required; (2) is there a *mathematical* method of showing the impossibility of the other value of C? To answer the second first. The other value of C from the above is

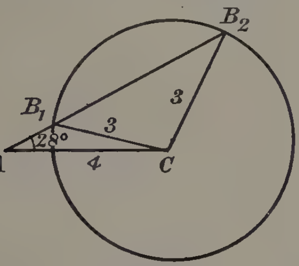

FIG. 80

$$180^\circ - 11^\circ\,46' = 168^\circ\,14'$$

when we should have

$$B = 180^\circ - (65^\circ + 168^\circ\,14') = 180^\circ - 233^\circ\,14'$$
$$= -53^\circ\,46'$$

Now a negative angle *in a triangle* is impossible, so that settles point (2). To settle point (1) we will work the following—

3. From A, a point 4 in. from the centre, C, of a circle of radius 3 in., a line is drawn making an angle 28° with AC. Find the distance of the points where it cuts the circle from A.

The data are represented in Fig. 80, where B_1 and B_2 are the points of intersection. In *both* triangles, AB_1C and AB_2C, we have $a = 3$, $b = 4$, and $\underline{/A} = 28^\circ$. Thus, the sine rule gives for them both

$$\frac{\sin B}{4} = \frac{\sin 28^\circ}{3}$$

or, $$\sin B = \frac{4 \sin 28^\circ}{3} = \frac{4 \times 0{\cdot}4695}{3}$$
$$= 0{\cdot}6260,$$

so that $$B = 38^\circ\,45' \text{ or } 180^\circ - 38^\circ\,45'$$
$$= 141^\circ\,15'$$

Referring to the figure it will be seen that the former of these is B_2, and the latter B_1, and the known properties of the isosceles triangle B_1CB_2 are in agreement with their being supplementary. Proceeding, we obtain two values for C, which are this time *both* possible.

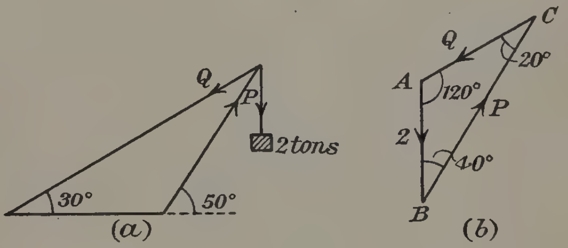

FIG. 81

C_2 (angle ACB_2) and C_1 (angle ACB_1)

$$C_2 = 180^\circ - (28^\circ + 38^\circ\,45') = 113^\circ\,15'$$

$$C_1 = 180^\circ - (28^\circ + 141^\circ\,15') = 10^\circ\,45'$$

By use of the sine rule again, we shall have two values of the side c, c_1 $(= AB_1)$ and c_2 $(= AB_2)$

$$c_2 = \frac{3 \sin 113^\circ\,15'}{\sin 28^\circ} = \underline{5{\cdot}871 \text{ in.}}$$

$$c_1 = \frac{3 \sin 10^\circ\,45'}{\sin 28^\circ} = \underline{1{\cdot}192 \text{ in.}}$$

4. If the radius of the circle were smaller, say, 1·5 in., the line would not cut the circle at all. What has the mathematical method to say about this case? Putting $a = 1{\cdot}5$ in the above, we find

$$\sin B = \frac{4 \sin 28^\circ}{1{\cdot}5} = 1{\cdot}2520$$

Now *no* angle can have a sine greater than 1, so that it is *impossible* to find the angle B; an impossibility corresponding to the fact that the line does not cut the circle. If a is greater than 4, the point A (Fig. 80) is within the circle, and the student should attempt to solve the question with such a value—though example (2) above is a similar case.

We give two further examples of the application of the sine rule, one to a mechanical problem (a type to which it is very often applicable), the other to a surveying problem.

5. The jib of a crane is at 50° to the horizontal, and the tie at 30°. The crane supports a load of 2 tons. Find the forces in the jib and in the tie.

Fig. 81 (*a*) shows the crane and load, the forces in the jib and tie being denoted by P and Q respectively. The triangle of forces is given in Fig. 81 (*b*), which is a triangle with one side (the vertical one) representing 2 tons, and angles of 40°, 120°, and 20°. Applying the sine rule to it, we have

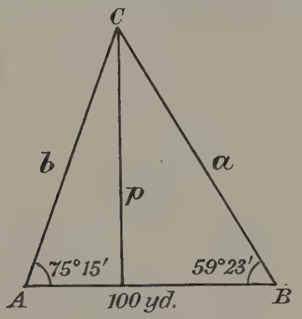

FIG. 82

$$\frac{P}{\sin 120°} = \frac{Q}{\sin 40°} = \frac{2}{\sin 20°}$$

whence

$$P = \frac{2 \sin 120°}{\sin 20°} = \underline{5{\cdot}064 \text{ tons}}$$

$$Q = \frac{2 \sin 40°}{\sin 20°} = \underline{3{\cdot}759 \text{ tons}}$$

6. From points A and B, 100 yd. apart on the bank of a river, a point C on the opposite bank is observed, and it is found that the angles CAB and CBA are 75° 15′ and 59° 23′ respectively. Find the width of the river.

The data are represented in Fig. 82, and we want the length of the perpendicular, p

Now $p = b \sin A$

and, by the sine rule,

$$\frac{b}{\sin B} = \frac{c}{\sin C} \quad \text{or} \quad b = \frac{c \sin B}{\sin C}$$

so that $$p = \frac{c \sin B \sin A}{\sin C}$$

In our triangle, $C = 180° - (A + B) = 45° 22′$ and $c = 100$.

Thus, $$p = \frac{100 \sin 59° 23' \sin 75° 15'}{\sin 45° 22'}$$

$$\doteq \underline{117 \text{ yd.}} \text{ (nearly)}$$

EXERCISE No. 29

1. In a survey, C being observed from A and B, $AB = 550$ yd., $\underline{/A} = 38°$, and $\underline{/B} = 85°$. Find AC and BC.

2. The crank of an engine is 3·5 in. long, and the connecting rod is 11·5 in. long. Find the position o the piston when the crank has turned through 50° an 60° from the inner dead centre position. If the spee of the engine is 900 r.p.m., find the average pisto speed in ft./sec. during this time.

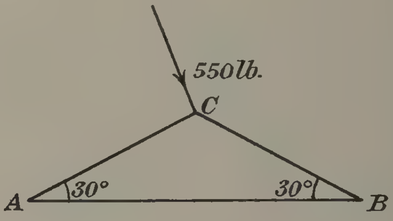

FIG. 83

3. ABC (Fig. 83) is a simple roof truss, the angle at A and B being each 30°. The weight of roof, an wind pressure combined, are equivalent to a force o 550 lb. acting at C at an angle of 10° to the vertica Find the forces in the members AC and BC under thi loading.

4. Show that the parts $a = 4$, $b = 2$, $B = 40°$ d not form a triangle, but that if $b = 3$ there are tw triangles. Find the length of the side c in each o these.

Cosine Rule. The next important rule fo triangles is the cosine rule, which we procee to demonstrate and apply. Referring to Fig 76, we will express the side c in terms of th other sides and the angle C.

In both figures, we have,

$$c^2 = p^2 + AD^2$$

and $$p = a \sin C.$$

Taking the figures separately to find AD, w have

(*a*)

$AD = b - CD$

$= b - a \cos C$

(*b*)

$AD = b + CD$

$= b + a \cos (180° - C$

$= b - a \cos C$

since $\cos (180° - C)$

$= -\cos C$

(*Continued on page* 880)

APPLIED MECHANICS

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LESSON XVIII

ELEMENTS OF MECHANISM

INTRODUCTION

Linear Motion. When we say that a body is in motion, we mean that it is changing its position, and usually we find it necessary to specify some other body relative to which the first body is changing its position. For example, when we refer to the motion of a piston, we mean that the piston is changing its position relative to the cylinder in which it works, that is, along the cylinder walls. Further, we may note that, if the cylinder is that of, say, a steam locomotive or a motor-car, which is itself in motion, then the whole motion of the piston is that which it has along the cylinder together with that which it has, because it is a part of the engine and is, therefore, travelling along the rails or road.

In any case, in our study of mechanisms, we shall find it convenient to state the point, relative to which the body under consideration may or may not have motion.

Linear Velocity. Since, ultimately, we are concerned with the *forces* which act upon the moving and fixed parts of a machine, we find it necessary to introduce the element of *time* into our study of the motion of a body, and in so doing we are led to conceive of a *time-rate of change* of a body's position. Let us suppose that at a certain time t, a body has moved a distance x from some fixed or zero position. During the next very small time interval δt, let its movement in a straight line be δx, then, the *mean* velocity over this time interval is $\frac{\delta x}{\delta t}$, and the actual velocity at the time t is the limiting value of this expression, which may be written—

$$Lt_{\delta x = 0}\frac{\delta x}{\delta t} = \frac{dx}{dt}.$$

We have so far considered velocity as being determined by the *two* quantities, *distance* and *time*, but we shall see later that there are really *three* quantities involved, viz., *distance, time,* and *direction.* If only *one* of these three quantities changes, there will be a change in the velocity. It is important to remember this, and we shall refer to it later when considering angular velocity.

Linear Acceleration. We are all aware that velocities may and do change, and that these changes require a longer or shorter time interval in different cases. This leads us to another notion, viz., the *time-rate of change of linear velocity*, which is what we mean by *linear acceleration.* Let us suppose that, at a certain time t, a body, which is moving in a straight line, has a velocity v, and that during the succeeding time interval δt, the velocity change is δv. The *mean time rate of change of velocity* or *linear acceleration* is given by $\frac{\delta v}{\delta t}$, and the *actual* linear acceleration at the time t is the limiting value of $\frac{\delta v}{\delta t}$, which may be written

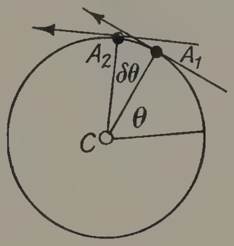

FIG. 98

$$Lt_{\delta v = 0}\frac{\delta v}{\delta t} = \frac{dv}{dt} = \frac{d^2x}{dt^2}$$

$$= f \text{ (the usual symbol).}$$

We must note that acceleration involves three quantities, viz., *velocity change, time, and direction.*

Circular Motion. The motion of bodies in a circular path occurs so frequently, that a study of circular motion becomes important.

Angular Velocity. Let A, Fig. 98, represent a body moving in a circular path about C as a centre; then, the *rate of turning* of the radius CA per unit of time may be taken as the angular velocity of A. We may state this with the aid of mathematical symbols as follows—

Let A_1, Fig. 98, represent the position of the moving body at time t, when it has moved through an angle θ, and A_2 the position of the body at time $t + \delta t$, when it has moved through the total angle $\theta + \delta\theta$; then $\frac{\delta\theta}{\delta t}$ is the mean angular velocity of the moving body during the time interval δt. The angular velocity of A at the time t may be expressed by $Lt_{\delta t=0}\frac{\delta\theta}{\delta t} = \frac{d\theta}{dt}$; which may conveniently be represented by ω. This angular velocity is usually stated in radians per second.

We have previously stated that the velocity of a moving body depends upon three quantities, viz., *distance*, *time*, and *direction*, and that a change in either of these meant a change in the velocity itself.

Let us consider a body, Fig. 99, moving uniformly in a circular path of radius R, with an angular velocity of ω radians per unit time. When the body is in position A, the magnitude of its linear velocity is ωR directed along the tangent to the circle at A. In some later position B, the magnitude of its linear velocity is still ωR, but it is now directed along the tangent to the circle at B. Clearly, there has been a change in the direction of motion, and this means a velocity change which can be determined as follows: Set down *ca*, Fig. 100, to represent ωR to any convenient scale and parallel to the tangent at A. Similarly, set down *cb* to represent in magnitude and direction the velocity of the body when at B, this again is ωR in amount. Join *ab*. We can now see that *ab* is the velocity which must be added *vectorially* to *ca* to give *cb* as the *vector* sum, hence *ab* represents in magnitude and direction the total *velocity change* undergone by the body in moving from A to B. Thus, we see that, although the *linear speed* of the body remains of constant magnitude, the *linear velocity* of the body is changing, this change being brought about by the change in the direction.

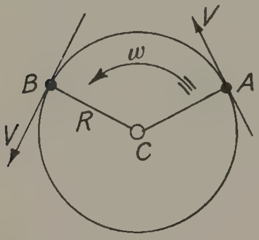

FIG. 99

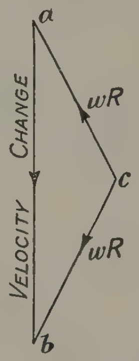

FIG. 100

Angular Acceleration. Just as we defined *linear acceleration* as the *time-rate of change of linear velocity*, so we define *angular acceleration* as the *time-rate of change of angular velocity*.

Let A_1 Fig. 101, be the position of a body, which at time t is moving with an angular velocity ω, and A_2, the position of a body, which at time $t + \delta t$, is moving with an angular velocity $\omega + \delta\omega$. Then, since the angular velocity change is $\delta\omega$, and the time change is δt, therefore $\frac{\delta\omega}{\delta t}$ is the *mean time rate of change of angular velocity*, and the *actual time rate of change of angular velocity*, that is, the *angular acceleration* at time t, may be written

$$Lt_{\delta t=0}\frac{\delta\omega}{\delta t} = \frac{dw}{dt} = \alpha \quad \text{(the usual symbol).}$$

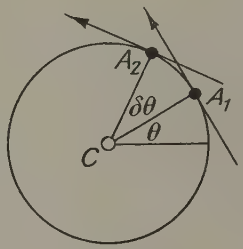

FIG. 101

EQUATIONS OF MOTION

To assist us in making our calculations relating to bodies having either linear or circular motion, the following equations of motion (with *uniform* acceleration) have been formulated—

LINEAR MOTION	CIRCULAR MOTION
$v = u + ft$ . . (1)	$\omega_1 = \omega_0 + \alpha t$. (4
$x = ut + \frac{1}{2}ft^2$ . . (2)	$\theta = \omega_0 t + \frac{1}{2}\alpha t^2$. (5
$v^2 = u^2 + 2fx$ . . (3)	$\omega_1^2 = \omega_0^2 + 2\alpha\theta$. (6

where
u = initial velocity
v = final velocity
f = linear acceleration
t = time of motion
x = movement during time t

where
ω_0 = initial angular velocity
ω_1 = final angular velocity
α = angular acceleration
θ = angular displacement
t = time of rotation

Derivation of Equation (1)

Since f means the change of velocity per unit of time, ft must mean the total change of velocity over a time interval t. If, now, the body possesses a velocity u before we begin to count time, then its final velocity must be the sum of its initial velocity and its acquired velocity; this may be stated as

$$v = u + ft \qquad . \qquad . \qquad (1)$$

Derivation of Equation (2)

If a body starts from rest with a constant acceleration f, then, in a time interval t, it will acquire a velocity ft. Since the body starts from rest, its average velocity during the time interval t is $\frac{ft}{2}$, and hence, the distance which it will cover in time t will be $\frac{ft}{2} \times t = \frac{1}{2}ft^2$. In the case of a body which already possesses a velocity u before we begin to count time, the average velocity is increased by this amount, and so becomes $u + \frac{ft}{2}$; hence, the distance which the body will move through in time t now becomes

$$\left(u + \frac{ft}{2}\right) \times t = ut + \tfrac{1}{2}ft^2 \qquad . \qquad . \qquad . \qquad (2)$$

Derivation of Equation (3)

This equation may be derived from (1) and (2).

Since $v = u + ft$, then by squaring we obtain

$$v^2 = u^2 + f^2t^2 + 2uft$$

or,

$$v^2 = u^2 + 2f(ut + \tfrac{1}{2}ft^2)$$

$$v^2 = u^2 + 2fx$$

(NOTE. $ut + \frac{1}{2}ft^2 = x$. See (2).)

Equations (4), (5), and (6) for circular motion can be derived from considerations similar to those for linear motion.

NEWTON'S LAWS OF MOTION

These laws, which are three in number, were known prior to Newton's statement of them, but they bear his name chiefly because he formulated them with great clearness and brevity.

Newton's First Law. "*If a body be at rest it will remain at rest; if a body be moving uniformly in a straight line it will continue in this uniform motion, unless acted upon by some external force.*"

It is impossible to demonstrate by experiment the truth of this law, because of the practical impossibility of freeing any moving body from the action of all external force. We may place our body on a surface which is level, to within very close limits of error, and so eliminate the effect of gravity, but there still remains the element of friction, and hence, since we cannot arrange a body to be free from all external force, we are unable to demonstrate how a body so freed would behave. Nevertheless, all our experience of force actions goes to support the statement embodied in the first Law.

Newton's Second Law. "*Rate of change of momentum is proportional to the impressed force, and takes place in the direction in which the force acts.*"

This law is of great importance since it provides us with a means of measuring force. We remember that there are three methods of measuring force, viz., we may note the change in the length of a spring when subjected to the action of a force; or, we may compare our force with the attractive force of the earth on a standard mass; or, we may measure the acceleration which our force produces when it acts upon a known mass.

Newton's Third Law. "*Action and reaction are equal and opposite.*"

This law, sometimes termed the *law of interaction*, can be demonstrated by causing two helical springs to be placed with their axes in line, and then to be loaded in, say, compression. It is obvious that each spring acts upon the other, and knowing the relative stiffness of the two springs and measuring the shortenings of the springs produced, we are able to demonstrate that each spring acts upon the other with an equal force and so establish the truth of the law. It is easy to show that the earth attracts the Newtonian apple, although it is impossible to show that the reverse is true; nevertheless, after performing the spring experiment as above, we have no hesitation in accepting the statement that the earth is equally attracted by the apple.

DERIVATION OF EQUATION—
$P = M \times f$

Newton's Second Law of Motion may be stated as—

Force $\propto$ time-rate of change of momentum

or force $\propto$ time-rate of change of (mass × velocity) or force $\propto$ mass $\times$ time-rate of change of velocity

$\therefore$ force $\propto$ mass $\times$ acceleration

or force $=$ a constant $\times$ mass $\times$ acceleration.

If, now, we correctly choose that force unit, which, when allowed to act upon unit mass produces unit acceleration, then we may write—

$$\text{Force} = \text{mass} \times \text{acceleration}$$

or, $$P = M \times f.$$

ABSOLUTE AND ENGINEERS' UNITS OF FORCE

From the above expression we can see that the unit of force will depend upon the units employed in our measurement of mass and of acceleration, whilst for the measurement of acceleration, we must employ the units of time and length. Now, the *British absolute unit of mass* is the *standard pound*, and the units of *time* and *length* are a *second* and a *foot* respectively, therefore, the absolute unit of force is that force which, acting upon a mass of 1 lb., gives to it an acceleration of 1 ft. per second per second. By experiment we find that a force of 1 lb. weight, acting upon a mass of 1 lb., produces an acceleration of "g" feet per second per second, or, a force of $\frac{1}{g}$ lb. weight, acting upon a mass of 1 lb. produces an acceleration of 1 ft. per second per second. This unit of force, $\frac{1}{g}$ lb. weight, is termed a *Poundal*, and because of its connection with the absolute unit of mass, is itself termed the *Absolute unit of Force.* Engineers, for practical reasons, prefer the larger unit of 1 lb. weight as the unit of force, and this requires their unit of mass to be g lb., hence, we may state the relationship as—

$$\text{Force (lb.)} = \frac{\text{mass (lb.)}}{g\left[\dfrac{\text{ft.}}{(\text{sec.})^2}\right]} \times \text{acceleration}\left[\frac{\text{ft.}}{(\text{sec.})^2}\right]$$

We can check the units as follows—

$$\text{lb.} = \text{lb.} \times \frac{\text{sec.} \times \text{sec.}}{\text{ft.}} \times \frac{\text{ft.}}{\text{sec.} \times \text{sec.}}$$

or $\text{lb.} = \text{lb.}$

MATHEMATICS FOR ENGINEERS

(Continued from page 876)

Thus, in both figures,

$$\begin{aligned} c^2 &= (a\sin C)^2 + (b - a\cos C)^2 \\ &= a^2\sin^2 C + b^2 - 2ab\cos C + b^2\cos^2 C \\ &= a^2 + b^2 - 2ab\cos C \end{aligned}$$

on collecting together the a^2 terms, and remembering that $\sin^2 C + \cos^2 C = 1$. Thus, our rule is

$$c^2 = a^2 + b^2 - 2ab\cos C \quad . \quad . \quad (B)$$

with, of course, similar formulae for b^2 and a^2. In this form it enables us to find any side if the other two and the angle between them is known.

It can be transformed into another useful form, a formula for the cosine of any angle in terms of the three sides, enabling us to find the angles of a triangle if we know the sides. Transposing the c^2 and the $2ab\cos C$ terms,

$$2ab\cos C = a^2 + b^2 - c^2$$

or, $$\cos C = \frac{a^2 + b^2 - c^2}{2ab} \quad . \quad . \quad (B')$$

with similar formulae for $\cos A$ and $\cos B$. Instances of the use of these formulae will be given in the next lesson.

ANSWERS TO EXERCISE No. 28

$+0{\cdot}7071$; $-0{\cdot}9659$; $+1{\cdot}7321$; $+0{\cdot}3420$; $-1{\cdot}1547$; $+0{\cdot}9848$; $+0{\cdot}8693$.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XVII

Reading Machine Drawings. We have, in a previous lesson, suggested that the way to acquire the art of reading machine drawings is to make drawings of many and various machine parts. Further, we have given a place in the art of machine construction to the reference drawing. We may now consider the use of the reference drawing, apart from its value in suggesting style, in the making of a new drawing.

Machine drawings often contain information which may prove invaluable to the draughtsman when employed upon the design of a machine part of which there is no close copy available. This information is not always on the surface of a drawing, but may be obtained with a little trouble. Often the information is comparatively worthless, but in other cases the potential data is invaluable. It is one reward of the art of reading a drawing with understanding that little time is wasted on the barren drawing. Given a little discrimination, the student can with practice choose only the most promising reference drawing for his analysis.

Potential Data. It may be added that when a knowledge of the behaviour of the machine part under working conditions is combined with the above-mentioned data, the information gained may prove as valuable as the results of a costly experiment. Further, we may remind the reader that the modern engineering periodical is very generous with detail drawings and particulars of the latest practice in machine construction.

Helical Gearing. As an example of the above method we may take the set of helical gearing shown in Fig. 1.

One of the first things to be observed in the above gear is the departure from the proportions, for straight toothed spur gears, mentioned in the last lesson. We may most readily find the reason for departing from the simple spur gear if we determine the sizes required for a straight toothed gear to transmit the power dealt with by the above set of helical gears. Let

T = twisting moment transmitted by the pinion in pounds-inches.

h.p. = horse-power transmitted by the pinion.

N = speed of pinion in revolutions per minute.

D = pitch circle diameter of pinion teeth in inches.

p = circular pitch of teeth in inches.

p_1 = normal pitch of teeth in inches.

W = total tangential tooth load in pounds.

w = total width of teeth under load in inches.

f = maximum stress induced in teeth in pounds per square inch.

θ = angle tooth makes with axis of rotation in degrees.

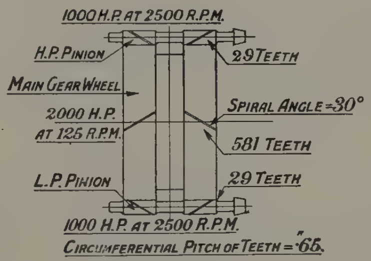

Fig. 1

Referring to Fig. 1, we have

$$T = \frac{63{,}000\,\text{h.p.}}{N} = \frac{63{,}000 \times 1000}{2500} = 25{,}200$$

lb. in., and, as was shown in Lesson XVI,

$$p = \left(\frac{T}{K^2.f.}\right)^{\frac{1}{3}} = \left(\frac{25{,}200}{1{\cdot}07^2 \times 4000}\right)^{\frac{1}{3}} = 1{\cdot}76 \text{ in.}$$

$$D = \frac{32 \times 1{\cdot}76}{\pi} = 18 \text{ in.},$$

and the diameter of the main wheel would be

$$= \frac{2500}{125} \times D = \frac{2500}{125} \times 18 = 360 \text{ in.}$$
$$= 30 \text{ ft.}$$

The above sizes are, of course, absurdly high, and to reduce them to a reasonable value it would be necessary to modify one of the limiting values used in determining the minimum tooth pitch. The stress f cannot, with safety, be increased, and if the tooth width w is increased beyond the previously acepted value of $3p$ we risk the danger of overstressing, due to the possibility of the tooth being end loaded.

Referring to Fig. 2, it will be seen that the teeth do not, as in the ordinary spur gear, run parallel to but at an angle to the axis of rotation. Further, the tooth running round the base circle at the above angle makes a helix, and gives to the pinion the name of helical gear.

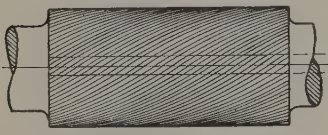
FIG. 2

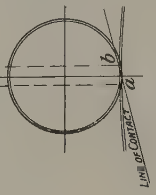

FIG. 3

It will also be observed that the tangential load W is taken by a number of teeth instead of one or two as in the case of the straight toothed gear. Further, each tooth is loaded over only a portion of its length, and so end loading is eliminated. Also the gradual engagement of the teeth, entering gear, gives a comparatively smooth running.

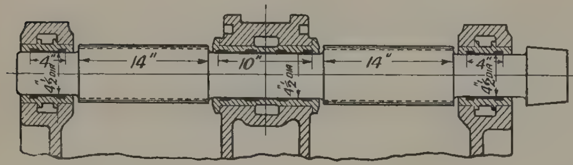

FIG. 5

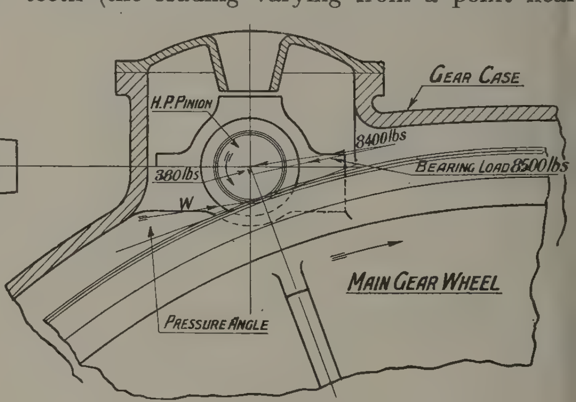

FIG. 6

It will be evident that, given a comparatively small tooth pitch, the total tooth length in contact is considerably greater than the limiting value of $3p$ as used for the straight toothed gear.

We may commence our analysis of the drawing by finding the length of the line of contact for each tooth in gear.

Referring to Fig. 3, the points of intersectio of the pinion and wheel tooth point circles wit the line of contact gives the length of contac ab. If we now draw (Fig. 4) a developed view of that portion of the pinion pitch circle cylinde within the length of contact ab, we may add lines, representing the tooth centre lines, a an angle of 30° to the pinion axis. The sum of the tooth lengths, as represented by the above centre lines, will give the equivalent length of tooth in gear. From this we may find the stress induced in the pinion teeth under load.

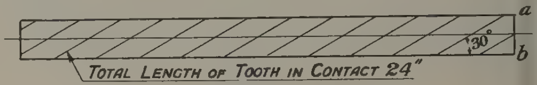

FIG. 4

$$\text{We have } W = \frac{T}{\frac{D}{2}} = \frac{25{,}200}{3} = 8400 \text{ lb.}$$

In finding the total bending moment on the teeth (the loading varying from a point near the root, at the approach, to the tip at the time of leaving contact) we may assume the load to be applied at the pitch circle. Then the approximate total bending moment on the

$$\text{teeth} = \text{B.M.} = \frac{W \times \text{tooth dedendum}}{\cos \theta}$$

$$= \frac{8400 \times 0{\cdot}25}{0{\cdot}866} = 2425 \text{ lb. in.}$$

and the total moment of resistance of the teeth.

$$\text{M. of R.} = \frac{w.d^2}{6} f = \frac{w.(\cdot 5\, K p_1)^2}{6} f$$

but, $p_1 = \cdot 65 \times 0\cdot 866 = \cdot 5628$ in.

then equating M. of R. = B.M.

$$f = \frac{6 \times 2425}{48\,(1\cdot 03 \times \cdot 5 \times \cdot 5628)^2} = 3608 \text{ lb./sq. in.}$$

$$\text{peripheral speed of teeth} = \frac{DN}{230} = \frac{6 \times 2500}{230}$$

$$= 65 \text{ ft.-sec.}$$

$$= 3900 \text{ ft.-min.}$$

Using the same variation of stress for different tooth speeds, as in the Lewis formula, we have substituting for f above

$$3608 = k\left(\frac{600}{600 + 3900}\right) = \cdot 133k$$

$$k = \frac{3608}{\cdot 133} = 27{,}130,$$

i.e. for any speeds sufficiently high to require a nickel steel pinion we may allow a value of $f = 27{,}130\left(\frac{600}{600 + V}\right)$ where V = tooth speed in feet per minute.

Maximum Tooth Load. It will be seen that, having dispensed with the limit for tooth width, some guidance is required in fixing a suitable diameter of pinion. The limit adopted, in lieu of the above $w = 3p$, is a maximum value of tooth load in terms of the pinion diameter D.

Let

P = tangential tooth load in pounds per inch width of face.

C = constant depending upon the type of gear (whether single or double reduction) and the service conditions.

Then $P = CD^{\frac{1}{2}}$ (empirical)

substituting from the above example

$$C = \frac{P}{D^{\frac{1}{2}}} = \frac{8400}{28 \times 6^{\frac{1}{2}}} = 122.$$

Bearing Pressure. Proceeding with the analysis we may find a working value for bearing pressure per square inch of projected surface.

Referring to Fig. 1 it will be seen that the gear being double helical, with the teeth converging to a point midway between the two faces, the axial components of the tooth load balance. The bearing load is the resultant of the tangential tooth load and the weight of the pinion. In calculating the pinion weight it will be sufficiently accurate to assume a cylinder of diameter equal to the pitch circle. From the dimensions given in Fig. 5 we find the weight of the pinion and shaft is 380 lb. Resolving the forces as shown in Fig. 6 we find the bearing load = 8500 lb.,

$$\text{bearing pressure} = \frac{\text{bearing load}}{\text{projected area of bearings}}$$

$$= \frac{8500}{18 \times 4\cdot 5} = 104 \text{ lb.-sq. in.}$$

$$\text{peripheral speed of journal} = \frac{4\cdot 5 \times 2500}{230}$$

$$= 49 \text{ ft.-sec.}$$

From the above we may assume a bearing pressure of 104 lb. per sq. in. of projected area at a speed of 39 ft.-sec. to be quite practicable for bearings having forced lubrication.

EXERCISE

Proceed with the analysis of the above example and find the constant k (in the Lewis formula) for the main gear wheel, using the value of K corresponding to the greater number of teeth.

THE THEORY OF HEAT ENGINES

By E. B. Cole, B.Sc.

LESSON XI

Rankine Cycle and Hypothetical Indicator Diagrams. In the last lesson we discussed the Rankine cycle for a liquid and its vapour, namely, water and steam, with the help of both the indicator or PV diagram and the $T\phi$ diagram.

Let us now see how closely this cycle is carried out in an actual reciprocating steam-engine.

The indicator diagram for an engine working on the Rankine cycle is reproduced in Fig. 28, in which, as we have seen, bc represents the *admission* of steam to the cylinder from the external boiler, the piston moving from b to c. Considering 1 lb. of steam, this represents the introduction of 1 lb. of steam of a given quality into the cylinder at a constant temperature T_1.

Adiabatic expansion then occurs, the piston moving to d and expansion being carried right down to the lower temperature limit T_2 at which the steam is to be condensed. *Exhaust* now takes place into the condenser to effect this.

The condensed steam is returned via a feed pump into the boiler, the pressure being raised by this means to P_1, and the temperature to T_1 by supplying sensible heat.

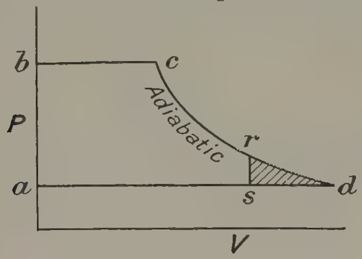

FIG. 28

Now in an actual engine the expansion is not usually carried right down to the condenser pressure, but is stopped at some point r, since it is found that below a certain pressure the work done by the steam in expanding further is less than that lost by engine friction and by condensation of the steam inside the cylinder; also, since the steam is finally condensed in a separate condenser, the valve should open to exhaust when the pressure inside is still above that of the condenser, so as to cause a flow of steam from the cylinder—a matter of importance in certain types of engine.

The drop in pressure to that of the condenser thus takes place at constant volume, and the modification is represented on the T-ϕ chart, Fig. 29, by the curve rs, which is a portion of the requisite "constant volume line."

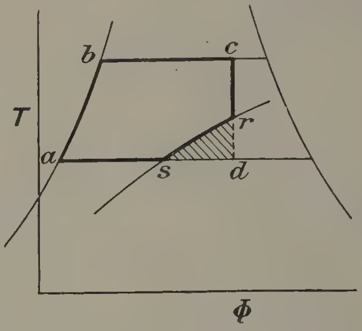

FIG. 29

The shaded area on both diagrams represents the work lost through this incomplete expansion.

In the indicator diagram, shown in Fig. 28, we have assumed that the piston comes right up to the end of the cylinder at the end of the exhaust stroke, but, in practice, a small *clearance* space must be provided for mechanical reasons, and at the same time it is usual for the exhaust port to close before the piston reaches the end of its stroke, so that some steam is trapped and compressed into the clearance space, thus giving a *cushioning* effect on the piston.

The hypothetical indicator diagram for an actual engine would be as shown in Fig. 30. At the point F the exhaust port closes, and compression occurs as represented by curve FA, while at A, the valve opens to steam and the pressure rises to that of the steam chest.

Again, in practice, the expansion curve will not be adiabatic, which would necessitate a perfectly non-conducting cylinder or, as an approximation, an isolated cylinder and a very high piston speed.

In a steam-engine fitted with an ordinary slide valve, each port is alternately used for admission and exhaust, so that when steam is admitted it flows into the clearance space and comes into contact with the cylinder walls, which will be at a relatively lower temperature, having just contained exhaust steam. Hence, if the steam were dry saturated on entry, condensation will occur as heat is transmitted to the cooler cylinder walls. The amount of this "initial condensation" will depend on the temperature difference between the steam and walls, the clearance volume and condition of its surface—that is, whether favourable to heat flow or not by being dry or wet, and the period during which the steam is in contact.

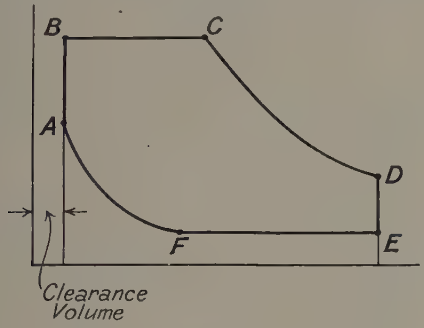

FIG. 30

Thus, during admission, heat is extracted from the steam and stored up by the cylinder walls, but towards the end of the expansion a point will be reached at which the saturation temperature of the cylinder contents will, due to the drop in pressure, fall below the temperature of the walls. Hence, there will now be a flow of heat from the walls, reducing their temperature and causing a certain amount of the condensed steam to re-evaporate. When release to exhaust occurs, due to the sudden pressure drop this re-evaporation takes place at an increased rate and continues during the exhaust stroke, the steam becoming about dry and saturated when compression starts.

This interchange of heat between the steam and the cylinder walls will obviously affect the shape of the expansion curve, and it is found that this curve approaches very closely to a rectangular hyperbola, the equation, therefore, being $PV =$ constant; so that, for all preliminary steam-engine design in which deductions have to be made from a hypothetical indicator diagram, the expansion curve, and also the compression curve, if this be taken into account, are assumed hyperbolic.

Let us, therefore, work out the theoretical mean effective pressure, taking into account the above modifications and assumptions, for the indicator diagram shown in Fig. 31, in which—

P_1 = initial pressure in pounds per square inch absolute.
P_B = final or back pressure per square inch absolute.
l = piston swept volume per stroke in cubic feet.
c = fraction of PSV forming clearance volume.
x = fraction of PSV over which compression occurs.
r = fraction of PSV forming admission volume.

Then—

$$V_C = (cl + rl) = l(c + r).$$
$$V_D = (cl + l) = l(c + 1)$$
$$V_F = (cl + xl) = l(c + x).$$
$$V_B = V_A = cl.$$

Work done on one side of piston by the steam = area $GBCDM$.

Work done by piston on steam = area $EMGA$.

Hence, net work done W = area of diagram.

$\therefore$ W = area $GBCK$ + area $KCDM$ − area $EMHF$ − area $FHGA$

$$= P_1(rl) + P_1 V_C \log_e \frac{V_D}{V_C} - l(1 - x)P_B - P_B V_F \log_e \frac{V_F}{V_A}$$

$$= P_1 l \left[r + (r + c) \log_e \frac{c + 1}{c + r} \right] - P_B l \left[(1 - x) + (c + x) \log_e \frac{c + x}{c} \right]$$

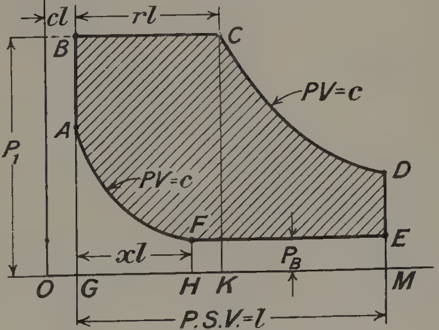

FIG. 31

Then the mean effective pressure in pounds per square ft. $= P_M = \frac{W}{l}$

$$\therefore P_M = P_1\left[r + (r + c)\log_e \frac{c + 1}{c + r}\right]$$
$$- P_B\left[(1 - x) + (c + x)\log_e \frac{c + x}{c}\right] \quad . \quad (1)$$

If there is no compression $x = 0$, then

$$P_M = P_1\left[r + (r + c)\log_e \frac{c + 1}{c + r}\right] - P_B \quad . \quad (2)$$

If clearance is also neglected $c = 0$, then

$$P_M = P_1 r\left[1 + \log_e \frac{1}{r}\right] - P_B \quad . \quad . \quad (3)$$

Total Heat—Entropy Diagram or Mollier Diagram. We must now discuss another diagram for a liquid and its vapour, first proposed by Dr. Mollier of Dresden, in which total heat I is plotted as ordinate against entropy ϕ as abscissa.

We have seen that the total heat or energy is given by

$$I = H + \frac{PV_0}{J}$$

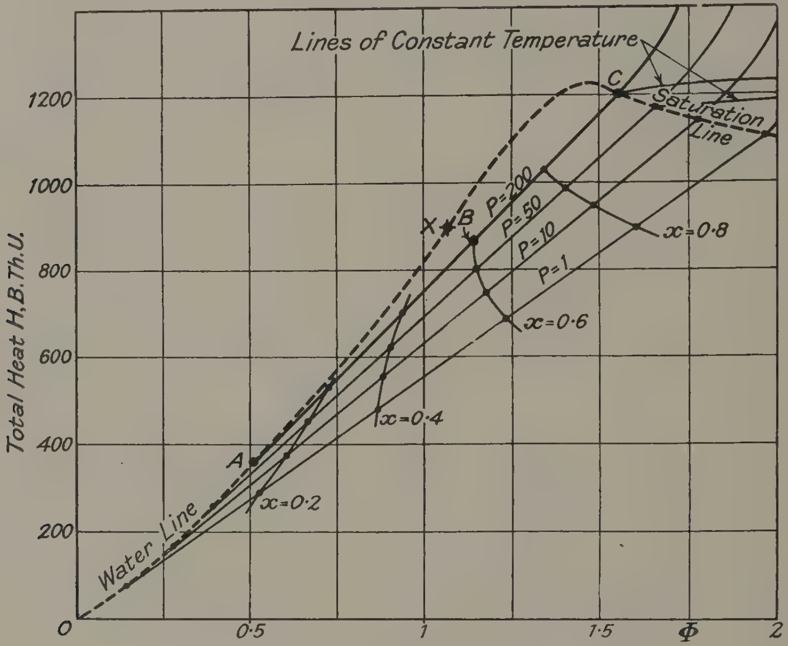

FIG. 32. MOLLIER TOTAL HEAT-ENTROPY CHART

and for purposes of ordinary calculation the latter term is so small as to be negligible, and the diagram would then become a total heat of evaporation – entropy chart.

The student is advised to make himself thoroughly familiar with this chart for water and steam by plotting the curves representing the heating of 1 lb. of water under different constant pressures, say, 1, 10, and 100 lb. per sq. in., obtaining the values of H and ϕ from the steam tables, or by direct caclulation as previously explained.

For instance, taking a constant pressure of 200 lb.-sq. in. absolute, we have—

1. *Water.* $H =$ sensible heat h

$$\phi_w = \log_e \frac{T_2}{T_1} \text{ approximately}$$

where $T_1 = (32 + 460)°$ F. absolute.

The saturation temperature corresponding to 200 lb. per sq. in. = 381·9° F.

The values of H and ϕ can thus be obtained for absolute temperatures increasing up to the saturation temperature, and for this latter temperature the steam tables give

$$H = 354{\cdot}9 \text{ B.Th.U.}$$
$$\phi_w = 0{\cdot}5437 \text{ units.}$$

Thus, on the chart, Fig. 32, which is to scale, OA represents the curve of relationship of H and ϕ for the heating of 1 lb. of water under a constant pressure 200 lb.-sq. in. up to the saturation temperature from (32 + 460)° F. absolute.

2. *Evaporation.*

$$H = h + xL$$

$$\phi = \phi_w + x\frac{L}{T}$$

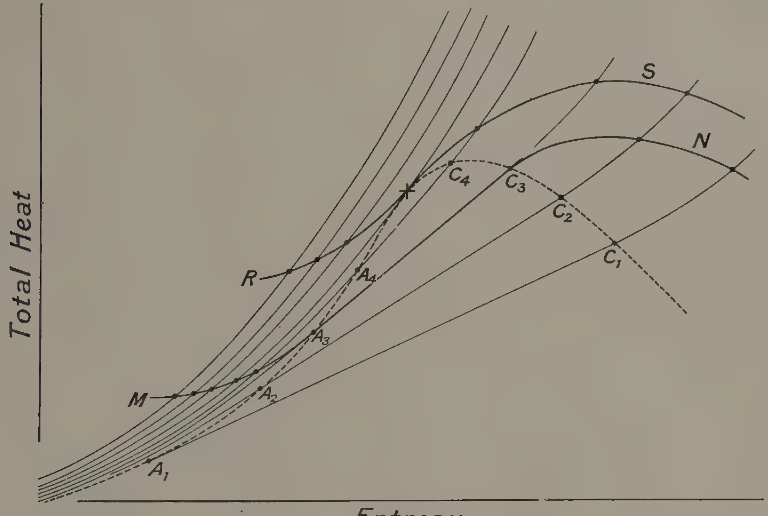

FIG. 33

Thus, for (say) a dryness $x = 0{\cdot}6$. taking steam table values

$$\therefore\ L = 843{\cdot}2$$
$$H = 354{\cdot}9 + {\cdot}6(843{\cdot}2) = 860{\cdot}8$$
$$\phi = 0{\cdot}5437 + {\cdot}6(1{\cdot}0019) = 1{\cdot}1448$$

where $1{\cdot}0019 =$ the entropy change during *complete* evaporation.

For dry saturated steam

$$H = 354{\cdot}9 + 843{\cdot}2 = 1198{\cdot}1$$
$$\phi = 0{\cdot}5437 + 1{\cdot}0019 = 1{\cdot}5456.$$

These results are represented on the diagram by the points B and C, so that the curve AC shows the relationship between H and ϕ during evaporation.

The diagram also includes similar curves for constant pressures of 1, 10, and 50 lb. per sq. in. absolute, and on joining similar points A we get a boundary curve called the "liquid line," while, joining similar points C we get the "saturation line." These curves are shown dotted and taken right up to the critical point. Compare the temperature entropy chart.

As with the $T - \phi$ chart, water being incompressible at ordinary temperatures, the curves showing the heating of water up to the evaporation temperature under any given pressures all coincide with the boundary curve when far removed from the critical, but on approach to this point, when the liquid becomes compressible, the constant pressure curves will separate out, as shown very much exaggerated in Fig. 33.

Again, since $I = H + \dfrac{PV_o}{J}$

when P is constant, then any change in I, say, $I_2 - I_1 = H_2 - H_1$, or for a very small change $\delta I = \delta H$.

Now the *slope* at any point of a constant pressure curve is

$\dfrac{dI}{d\phi}$ which is, therefore, equal to

$\dfrac{dH}{d\phi}$, but, from the definition of entropy,

$$d\phi = \frac{dH}{T}$$

$$\text{or}\quad T = \frac{dH}{d\phi}$$

we thus see that the slope at any point of a constant pressure curve gives the absolute temperature of the substance under the conditions represented.

Thus, the slope of the constant pressure curve 200 lb.-sq. in. at $A = (381{\cdot}9 + 460)^\circ$ F. absolute, and from A to C, representing evaporation at this constant temperature, the curve is therefore a straight line. From C onwards in the superheat region the slope increases continuously as the temperature rises.

The critical point in Fig. 32 is not at the apex of the boundary curve, but about at the point X, which will be the point of maximum slope of the boundary curve, the constant pressure curve for the critical pressure just touching the boundary at this point. Thus, the slope of the boundary curve at X is equal to the critical temperature.

On the complete chart corresponding points representing the same dryness are joined up, giving *lines of constant quality* as shown in Fig. 32, while in the superheat region points on each constant pressure curve having the same slope are connected, giving *lines of constant temperature.*

With a compressible liquid, i.e. in the neighbourhood of the critical point, constant temperature lines are shown in the liquid region, as indicated in Fig. 33. Two constant temperature curves are here shown ; the firs MA_3C_3N, is drawn through points on eac constant pressure curve that have the sam slope as the point A_3 on its constant pressur curve i.e. the curve represents the expansion c the substance at a constant temperature equal t that of saturation corresponding to the pressur for which the curve A_3C_3 is drawn, sinc evaporation is about to start at the point A MA_3 represents expansion of the liquid at thi temperature, A_3C_3, coinciding with the constant pressure curve, indicates evaporatio occurring at this temperature, and C_3N superheating of the vapour at constant temperatur but falling pressure.

The second curve RXS is drawn throug points having the same slope as the critica pressure curve at X, i.e. represents a constan temperature equal to the *critical temperature.*

The portion RX is a boundary line—below i and outside the dotted liquid line the substance can only exist as liquid ; above it and outside the saturation line vapour is the only possible state. Compare again the T–ϕ chart.

The PV diagrams corresponding to these constant temperature changes will be similar to the isothermals previously described for CO_2.

THE TESTING OF PRIME MOVERS

By R. Royds, M.Sc., M.I.Mech.E.

LESSON III

THE MEASUREMENT OF PRESSURE

The measurement of pressure is so common a feature in the operation of power plant that it is hardly necessary to enumerate here the pressures to be measured under test conditions. The method of measurement to be adopted depends largely upon the character of the pressure, that is, whether the pressure is small or large, and whether steady or fluctuating. To some extent the method adopted in a particular case may depend also upon the degree of accuracy required. In all cases it is a difference of pressure that is measured. If the pressure is measured or calculated from the zero of pressure it is said to be an absolute pressure, and if measured from and above atmospheric pressure it is commonly referred to as a gauge pressure. A pressure measure from and below atmospheric pressure is terme a “ vacuum.” The most accurate means o measuring a steady pressure, or one that i subject only to minor fluctuations, is to balanc the pressure against a liquid column. Fo obvious reasons such a method under ordinary conditions is limited to relatively smal pressures.

Barometer. The most common example i that of measuring atmospheric pressure by th mercury barometer. The principle of construction is represented in Fig. 8. It consists of glass tube T, closed at the top end, and with th open end in a vessel of mercury V. The tub is originally filled with clean mercury, free fro water and bubbles of air, with the open en upwards, and is then upended and lowered int the mercury in the vessel. The pressure ove the mercury in the closed tube is that of th vapour of mercury, and is so small as to be practically negligible at ordinary temperatures. A pointer *P* is provided at the zero level in the vessel, and in the best instruments the bottom of the vessel is flexible, so that the level in the vessel can be adjusted to the zero point by a screw *S* acting on the flexible leather bottom. The height of the mercury then represents the pressure of the atmosphere, and the reading should be taken at the top of the column in the centre of the tube. A vernier scale is commonly provided to increase the accuracy of the reading if that should be required.

Vacuum Gauges. A column of mercury is commonly used to measure the vacuum in a condenser. Such an arrangement is shown in Fig. 9. In principle it is similar to a barometer, except that the top end of the tube is connected to the condenser through a cock and a copper pipe. The scale should be adjustable so as to place the zero on the scale at the level of the mercury in the bottle or vessel. If the scale is fixed, then the divisions on the scale need to be smaller than the readings on a true scale, to allow for the fall of level in the bottle as the mercury ascends the tube. The copper pipe leading from the top of the tube should be arranged so that water cannot accumulate in the pipe, and the cock should be kept closed when readings are not required. Notwithstanding these precautions water vapour sometimes condenses and accumulates on the top of the mercury in the column. The reading of the mercury column should then be increased by the amount equivalent to the water column according to the relative densities, that is, divide the height of the water column by 13·6 to get the equivalent height of mercury. Some of the water may be got rid of by detaching the pipe and blowing the water into the vessel of mercury. A better method is to hold a piece of hot metal close to the tube, but not touching it, to evaporate the water in the tube. When all the water has been thus evaporated, allowing a small amount of air to be drawn into the pipe connection at the cock at the top of the column, will replace the vapour in the pipe by air almost free from vapour.

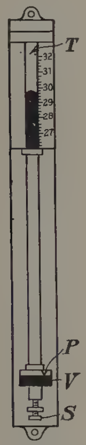

FIG. 8. MERCURY BAROMETER

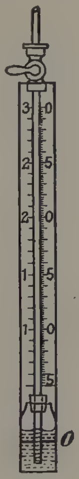

FIG. 9. VACUUM COLUMN

The absolute pressure in a condenser is represented by the difference between the readings of the barometer and the vacuum column. This absolute pressure may be obtained directly by the arrangement shown in Fig. 10, where the closed tube is originally filled with mercury. When the cock is opened the difference of level *h* represents the absolute pressure on the surface of the mercury on the right-hand column, and hence that in the condenser. It is not an easy matter to get the closed end of the tube completely filled with mercury and free from air. If the glass tube is fairly wide in bore a small rubber tube may be passed down to the closed end and the mercury poured through this rubber tube, gradually withdrawing it as the glass tube fills. The relation between inches of mercury vertical column and pounds per square inch is given for ordinary temperature conditions by multiplying inches of mercury by 0·49.

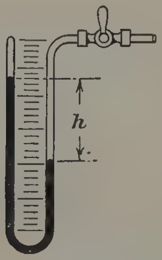

FIG. 10. ABSOLUTE PRESSURE COLUMN

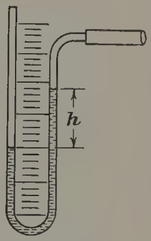

FIG. 11. WATER MANOMETER

Water Gauge. Very small differences of pressure are usually measured by a column of water commonly called a water manometer. Such small pressures occur due to the draught of a chimney or the pressure difference created by a fan. An ordinary construction of water manometer is shown in Fig. 11. A glass tube is bent to the form shown, the diameter of the tube being not less than $\frac{1}{4}$ in. say, and is clipped to a board with a scale behind. A convenient scale is made by pasting white paper on the board and when dry marking the scale in inches and decimals of an inch, afterwards varnishing it over to keep it from becoming soiled. The glass tube can then be fixed over the scale. When the pressure difference is very small, as when measuring a chimney draught, some difficulty may be experienced in reading the water columns in the two legs. It may then be more convenient to place the tube with the board on a definite slope, as represented in Fig. 12, in elevation and plan. This widens the scale reading in the relation of unity to the sine of the angle of slope, but the arrangement may be calibrated by comparing with a vertical column. A level is required on the base board with one or more levelling screws to make certain that the tube is set at the correct slope before taking readings. Oil may be used instead of water if the density of the oil is known.

The resistance offered to the steady flow of water through a pipe or through a condenser may be measured by columns of water in an upended **U** tube as shown in Fig. 13, where the tube is shown connected to the inlet and outlet of the circulating water system in a surface condenser. The leg A_1 is connected to the water inlet and leg A_2 to the outlet by means of stout rubber tubing or hose. These connecting tubes have to be full of water, and the cock B enables the level of the water in the tubes $A_1 A_2$ to be adjusted by introducing air under suitable pressure. In a small condenser where the water at outlet flows away freely at near atmospheric pressure, a sufficient pressure at B may be obtained by blowing into th system at B with the mouth, but otherwis bellows may be necessary. The difference of lev h represents the resistance offered to the flo of the circulating water through the condense

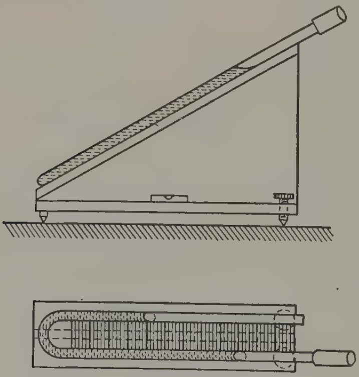

Fig. 12. Water Manometer

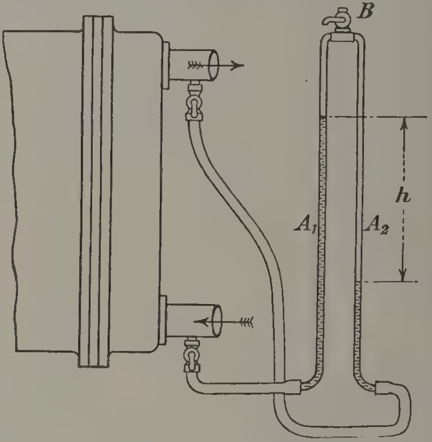

Fig. 13. Arrangement of Water Column Connected to Condenser

The length of the tubes A_1 and A_2 required i a large high speed condenser may be too grea for convenient reading. In that case th arrangement shown in Fig. 14 may be adopted where the lower end of the **U** tube is filled wit mercury and the two legs connected to th inlet and outlet as before. The difference o level h then represents the resistance; but i should be remembered that it is in terms of th difference between the density of mercury an water. The swelled portions of the tubes nea the top are introduced as a precaution agains the mercury being shot out of the **U** tub should the pressure come on suddenly.

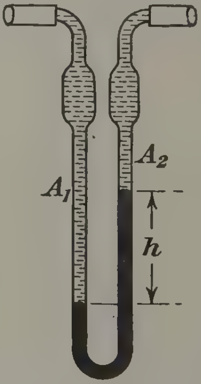

Fig. 14. Mercury Column for Measuring Pressure Difference

Several makers can supply manometers of finger and dial type, and also recorders to show the pressure differences on a chart to a basis of time.

Pressure Gauges. A pressure gauge is a common arrangement for measuring "steady" pressures beyond the convenient range of liquid columns, or even for small pressures where an extreme degree of accuracy is not required. The usual construction is represented in Fig. 15. A steel or bronze tube *B*, of oval section as shown at *S*, and known as a Bourdon tube, is brazed to the brass connection *A* and is blocked up at the end *E*. When subjected to an internal pressure above that of the atmosphere the tube dilates and the end *E* moves outwards. The motion of *E* is too small to indicate the pressure directly, so it is connected to some form of multiplying gear. This gear usually consists of the link *L*, connected to the slotted end *D* of a quadrant, which is pivoted at *O*. The teeth *Q* on the quadrant gear with the pinion *P* and on the same spindle is the pointer or finger *F*. Thus when the end *E* moves it causes the pointer to move over the scale of pressure round the dial. A light coiled spring is sometimes attached to the spindle of the pointer so as to keep the teeth on one side always in contact, thus taking up any slack in the teeth, whether there originally or due to wear.

When a steam pressure is to be measured the hot steam needs to be kept out of contact with the tube because the heating of the tube affects its elasticity. For this purpose a siphon tube connection would intervene between the steam and the gauge, filled with cold water or with condensed steam, which eventually becomes relatively cold if no leakage occurs. A shut-off cock would also be introduced, so that the gauge can be detached if necessary.

Vacuum gauges are similarly constructed, the tube contracting when subject to pressures below atmospheric pressure.

Gauge Errors. Like most instruments, gauges are subject to errors of the following character: (1) a constant error; (2) errors which change more or less gradually over the range of pressure; (3) discrepancies between the readings obtained with rising pressures as against those given with falling pressures; (4) erratic errors which do not seem to follow any law of change.

So much faith is sometimes placed in the readings of a pressure gauge that it is perhaps advisable to discuss further the possible causes of error, and to describe the methods by which gauges are tested. Special test gauges are built by most gauge makers for the use of boiler inspectors and the like. These gauges are specially constructed so as to try to eliminate all sources of error, and usually have two tubes with two separate mechanisms controlling two pointers on the same centre, so that one is a check against the other. Special connections are provided, so that both the test gauge and the gauge to be tested can be connected to the same pressure, say the boiler pressure. Even a test gauge cannot always be relied upon. On several occasions the author has examined test gauges for error due to having been accidentally let fall, and in some cases has found a practically constant error has developed. The best method of testing a gauge is to compare its readings at known pressures, both with pressures rising and falling.

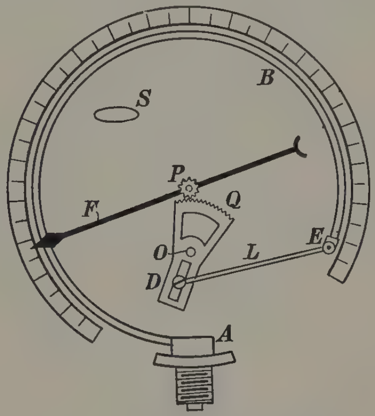

Fig. 15. Pressure Gauge Mechanism

Gauge Testers are available for this purpose, the simplest form being that shown in Fig. 16, made by the Budenberg Gauge Co., Ltd. A weighted plunger passes through a bush in the stand pipe shown, which is in communication with the hollow base. The gauge is screwed on to the cock and is thus subjected to the definite pressure created by the load on the plunger acting on oil in the base and stand pipe. Before a reading is taken the plunger with its load is gently spun round to eliminate friction between the plunger and the bush. The cup at the top of the stand pipe contains the oil which may leak past the plunger during a test. If the gauge in use is subject to a small pulsation of pressure or vibration the gauge, when on test, may be tapped before reading it. Otherwise the gauge would not be touched. In common gauges there is sometimes an appreciable difference between the readings obtained under these two conditions.

If the error found is practically constant it may be corrected by forcing the pointer off its conical fit on the spindle and replacing it in the correct position with a definite pressure on the gauge, but not at zero pressure. An error that gradually changes as the pressure rises can be corrected by shifting suitably the point of connection *D* in the slot in the quadrant (Fig. 15) until the error is constant, and then the pointer would be moved as previously mentioned. Discrepancies between readings as between rising and falling pressures is usually due to frictional resistance in the mechanism or due to some slackness. Watch should be taken to see that the pointer is not touching the dial. When a gauge has been badly used, as, for instance, being accidentally subjected to a pressure much in excess of its range, the elasticit
of the tube might be affected and thus introduc
errors into the gauge reading, probably erro
of an erratic nature. The elasticity of the tub
and the position of the pointer on the piv
might also be affected if the gauge has bee
subjected to pressures which were rapidl
fluctuating over a long period.

Vacuum gauges may be tested by connectin
a vacuum column, constructed as shown i
Fig. 9, to the same pressure system as th
gauge, or a special tester may be used, consist
ing of a vacuum column and a hand air pum
to which the gauge may be connected fo
comparison.

Recording gauges are also available where th
gauge mechanism previously described is mod
fied so as to allow the inked pointer to draw
pressure line on a moving chart driven b
clockwork. These are useful when it is desire
to have a check on the manner in which th
boiler or other pressure is controlled.

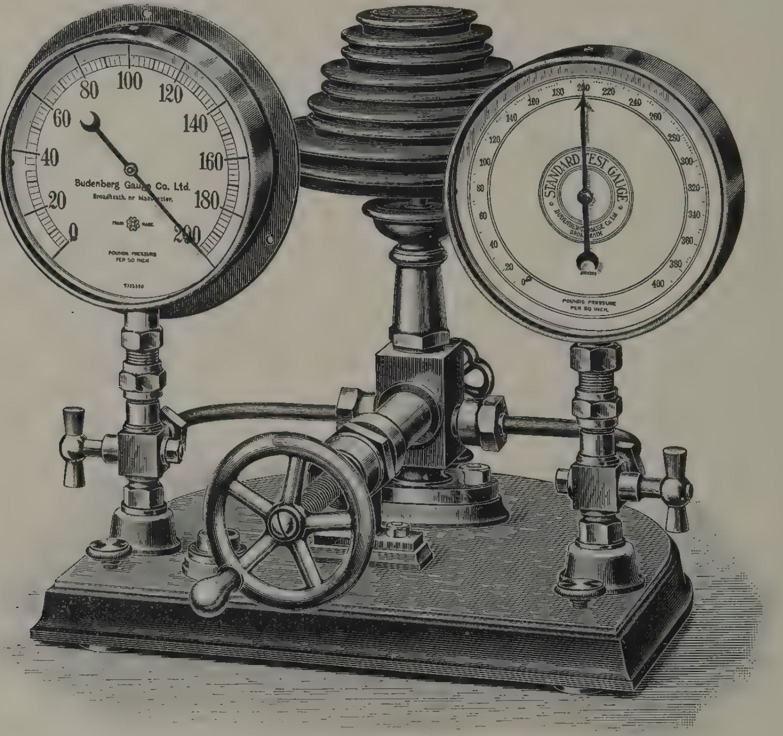
Budenberg Gauge Co. Ltd.
Broadheath or Manchester.
POUNDS PRESSURE PER SQ INCH
0
20
40
60
80
100
120
140
160
180
200
STANDARD TEST GAUGE
POUNDS PRESSURE PER SQ INCH
400

FIG. 16. PRESSURE GAUGE TESTER

REFRIGERATING MACHINERY

By G. W. Daniels, M.Eng., Wh.Ex., A.M.I.Mech.E.

LESSON I

INTRODUCTION

Refrigeration is one of the most fascinating and interesting of all branches of engineering. This great interest is largely due to the fact that refrigeration is not solely a matter of engineering, but wanders into the realms of the majority of other sciences, even to bacteriology, bio-chemistry, etc. A successful refrigerating engineer must, therefore, have some working knowledge of these sciences, although naturally he cannot be an expert in them all, and the diversity of subjects with which he must be to some extent familiar, creates a breadth of view and interest seldom equalled and never surpassed by any other branch of engineering.

The subject is also of the highest importance to every member of a modern community, as there are few industries which do not require the aid of refrigeration at some stage in the products which they produce. Thus we depend on refrigeration for preservation of our food supplies, for the manufacture of medicines, certain wearing apparel, and a multitude of other things too numerous to detail.

Furthermore, our science of refrigeration is of very ancient origin. It is on record that Alexander the Great made use of snow to preserve wines, and in tropical countries, such as India, a crude method of refrigeration has been in use from the earliest times, and is even to-day employed for household use in an improved form. This ancient device consisted in exposing water, contained in shallow porous vessels, to the night wind. Some of the water soaked through the porous vessel to the outside and was there evaporated by the wind. The evaporation of the water in this way abstracted heat from the remainder and cooled it.

The fact that all liquids take up heat when they evaporate is the underlying principle of practically all modern refrigeration. Many instances of this fact will readily occur to the reader, e.g. the water in a boiler abstracts heat from the furnace gases as it evaporates into steam.

Elementary Refrigerating System. Fig. 1 shows diagrammatically the arrangement of a refrigerating plant in which the cold is produced by the evaporation of a liquid.

A is a coil of pipe, immersed in a tank, *B*, which may contain the substance we desire to cool. The liquid which is to produce the cold is allowed to evaporate inside the coil of pipe, and in consequence it abstracts heat from the substance outside the pipe, and so cools it. The liquid which evaporates and produces the cold is termed the "refrigerant." As the liquid refrigerant evaporates it turns into a vapour, and this vapour is removed from the coil of pipe *A* by a pump or compressor *C*. The coil *A* is called the refrigerator or evaporator. The compressor or pump *C* forces the vapour of the refrigerant into a second coil *D*, immersed in water and called the condenser. In this the vapour is turned into a liquid again by the joint action of the pressure produced by the pump *C*, and the water surrounding the condenser. The liquid refrigerant then passes from the condenser, through a regulating valve *E* back into the refrigerator, or evaporator, where it again evaporates and produces more cold.

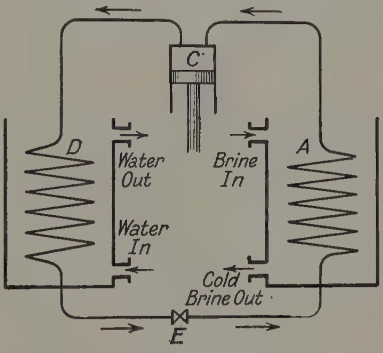

Fig. 1. Diagrammatic Arrangement of Refrigerating Plant

From a consideration of this elementary refrigerating system, several important and interesting features emerge.

Firstly, we see that the refrigeration is produced solely by the evaporation of the refrigerant in the evaporator. If we allowed the vapour from the evaporator coil to escape into the atmosphere instead of removing it by the compressor, we should still produce as much refrigeration as before. We should, however, lose our refrigerant, and would continually be purchasing fresh supplies. The compressor or refrigerating machine itself, therefore, does not produce the refrigeration. It simply removes the vapourized refrigerant from the evaporator instead of allowing it to escape into the atmosphere, and delivers it into the condenser where it can be liquefied for further use. The main function of the compressor and condenser is, therefore, to render the refrigerating process a commercial one.

We further see that the refrigerant passes continuously round and round the system, or as we say, through a cycle of operations.

Again, it is obvious that by regulating either or both the speed of the compressor which is removing vapour from the evaporator, and the quantity of new liquid passing into the evaporator, we can vary the pressure inside the evaporator coils, and thus alter, at will, the temperature at which the liquid refrigerant evaporates, and so vary the low temperature produced. All engineers will be familiar with the fact that the temperature of evaporation varies with the pressure, the most familiar example being the variation of the evaporating temperature of water with the pressure in a steam boiler.

The refrigerating process is thus governed by three simple physical laws or facts—

1. That any liquid when evaporating absorbs heat from its surroundings.

2. That all vapours can be changed to the liquid state by a suitable combination of pressure and cooling.

3. That the evaporating temperature of a liquid depends upon the pressure under which it evaporates.

The first law makes it possible to produce refrigeration, the second makes the refrigerating process a commercial success by permitting us to use the same refrigerant over and over again, while the third law enables us to control the degree of refrigeration, that is, the low temperature produced.

Refrigerating Machine a Heat Pump. In all refrigerating problems we desire to lower the temperature of some object or substance below that of the surrounding atmosphere. Now the second Law of Thermodynamics tells us that it is impossible for heat to flow from a low temperature to a higher temperature by itself without assistance; in fact, in order to effect such a transfer we must expend some energy on it, and force the heat from a lower temperature to a higher temperature. This gives us the idea of the refrigerating machine being a *heat pump*, which pumps heat from the low level of temperature of the refrigerated body and carries it to a higher level of temperature. What becomes of the heat pumped out of the refrigerated body? Considering the analagous case of pumping water out of a mine, having got the water to the surface, we let it run to waste in a river, stream, drain, in fact anywhere where it can be carried off and not trouble us any more. The same applies to our refrigerating machine. Having pumped the heat to a higher level of temperature we let it run to waste and it is, in fact, carried away by the condensing water surrounding the condensing coils.

If H = amount of heat carried away by the condensing water,
h = amount of heat removed from the body to be refrigerated,
w = work expended in driving the refrigerating machine, expressed in heat units,
then $H = h + w$.

As a measure of the efficiency of a refrigerating machine we take the amount of refrigeration produced for each unit of power expended in driving the machine. We usually call this quantity the *coefficient of performance*. In symbols—

$$\text{coefficient of performance} = \frac{h}{w}$$

both h and w being expressed in heat units.

EXAMPLE. A machine produces refrigeration equal to 40,000 B.Th.U.'s per hour, and takes 13 h.p. to operate it. What is its coefficient of performance?

$$13 \text{ h.p.} = \frac{13 \times 33{,}000 \times 60}{778} = 33{,}100 \text{ B.Th.U.'s per hour}$$

$$\text{Coefficient of performance} = \frac{40{,}000}{33{,}100} = 1{\cdot}21.$$

A theoretically perfect refrigerating machine would have its coefficient of performance given by the expression—

$$\frac{T_2}{T_1 - T_2}$$

where

T_2 = temperature of the refrigerator
T_1 = temperature of the condenser.
both measured on the absolute scale.

This shows us that to get the greatest coefficient of performance, that is, the most economical refrigerating machine, we should not make T_2, the temperature of the refrigerator, any lower than is absolutely necessary for the work in hand, and further, T_1 should be kept as low as possible by using a good supply of the coolest condensing water available.

The coefficient of performance of any actual refrigerating machine is always, of course, less than that of the ideal machine, and in both the ideal and actual machines the coefficient varies greatly with the working conditions.

Unit of Refrigeration. Refrigeration being a science dealing with quantities of heat, then any of the usual heat units, British thermal unit, calorie, etc., can be used for denoting quantities of refrigeration.

There has grown up with the industry, however, another unit which has some practical convenience. This unit is called the *Ton of Refrigeration*, and its magnitude is equal to the amount of heat necessary to melt 1 ton of ice in 24 hours.

Thus in heat units,

One ton refrigeration

= 2,240 lb. × 143 B.Th.U.'s
= 322,000 B.Th.U.'s per 24 hours
= 13,000 B.Th.U.'s per hour
= 224 B.Th.U.'s per minute.

In America the value of the ton of refrigeration is about 10 per cent less than in this country, because the American ton is one of 2,000 lb. only. Care must, therefore, be taken to distinguish between English and American tons of refrigeration.

The usefulness of the ton of refrigeration lies in the fact that it is roughly equal to the refrigeration required to freeze 1 ton of miscellaneous cold stored goods per day under average working conditions in this country, and if the machine is employed in making ice, the amount of ice produced is approximately equal to one-half the capacity of the machine expressed in tons refrigeration.

REFRIGERANTS

Ammonia. Theoretically it does not matter what working substance or refrigerant we use to produce the refrigeration, but in practice it matters a great deal, and we therefore find that actual refrigerating machines use one or other of a few refrigerants which experience has shown to give satisfactory results.

The most widely used refrigerant at the present day is ammonia. This is a chemical compound of nitrogen and hydrogen, denoted by the chemical symbol NH_3. Ammonia has a penetrating odour which enables small leaks to be detected; is colourless; attacks copper and copper alloys such as brass, which therefore cannot be used in the construction of ammonia refrigerating plants; the vapour is lighter than air and the liquid is lighter than water. It is not inflammable, although under certain conditions it can be made to burn. The pressures required to liquefy the ammonia in the condenser are moderate, seldom exceeding 150 lb. per sq. in. in this country. One pound of liquid ammonia can absorb quite a considerable amount of heat, and therefore, can produce a considerable amount of refrigeration. The amount of ammonia vapour required to be handled by the compressor is moderate in any given case. Ammonia is poisonous if it escapes into the atmosphere in large quantities.

Carbon Dioxide. The next most widely used refrigerant is carbon dioxide, or CO_2. This substance is also colourless, but has no smell. The vapour is considerably heavier than air. It is not poisonous if it escapes into the atmosphere, but if there is an excessive quantity present in a confined space it may produce suffocation. One pound of liquid CO_2 will only produce a small amount of refrigeration, but to counterbalance this the amount of vapour to be handled by the compressor is extremely small and, therefore, only a small compressor is required. The pressure in the condenser is very much higher than with other refrigerants, and may reach 1,500 lb. per sq. in. CO_2 is the most extensively used refrigerant for marine work on account of its non-poisonous properties.

Sulphur Dioxide. Another refrigerant, old established, but now seldom used on a large scale, is sulphur dioxide or SO_2. Sulphur dioxide has a pungent smell and is poisonous if present in the atmosphere in large quantities. It does not attack any of the metals used in machine construction if it is dry, but may do so if it becomes wet. One pound of liquid SO_2 can only produce a small amount of refrigeration, and the volume of vapour to be handled by the compressor is very large, thus necessitating a large machine. This is a serious disadvantage. In fact, the only advantage which an SO_2 machine may claim is that the operating pressures are low. In some instances these low pressures may be a positive disadvantage.

BLAST FURNACE PRACTICE

By Percy C. R. Kingscott, D.I.C., A.R.C.Sc., F.I.C., B.Sc. (Lond.)

LESSON VIII

PLANT—(*contd.*)

Water Supply

A blast furnace plant requires for its cooling members, gas washing, boilers, etc., a large quantity of water. Generally, cooling water used in coolers, tuyeres, hearth jackets, etc., is in continuous use, being allowed to cool in suitable ponds usually provided with water sprays. Any evaporation taking place is then made up either continuously or intermittently. In order to provide sufficient head, viz., 20–25 lb. to the square inch, the water is pumped into a tower tank of sufficient dimensions to carry on the plant for a short time in case of pump troubles.

Owing to the difficulty in disposing of effluent water, that issuing from gas cleaning plants is generally treated in settling ponds, the clear water being used again. In this connection, the Dorr thickener has proved useful. The recovered sludge is sometimes incorporated into the furnace burden; e.g. the thick sludge from settling tanks is pumped continuously to a ring distributor under the furnace bell and spread over the burden in a furnace at the Clairton Works (U.S.A.), the system being described in the *Iron and Coal Trades Review*, May, 1921.

Pumping is usually done by centrifugal pumps electrically operated and situated in central pumping stations. It is wise to have a pump driven by a steam-engine as a standby and, owing to the extreme danger to the plant caused by water failure, it is absolutely essential that a constant supply be ensured.

The furnace is encircled, usually in the neighbourhood of the bustle pipe, with a cast water pipe joined by service main to the tank, this supply being controlled by a valve. The circular pipe is in turn connected to each cooling member on the furnace, the waste water being led away usually to a common discharge channel.

Gas Requirements, etc., of a typical furnace.

Let it be supposed that a furnace be required to make 500 tons of pig per 24 hours, with a coke consumption of 20 cwt. per ton of pig.

Assuming that 5,500 lb. of standard coke are burned per square foot of hearth area per 24 hours, then the area of the hearth is—

$$\frac{500 \times 2240}{5500} = 204 \text{ sq. ft.}$$

which is equivalent to a diameter of about 16 ft.

Assuming that it requires 60 cub. ft. of air, including losses, to burn 1 lb. of coke, the volume of air at the blowing engine is—

$$\frac{500 \times 2240 \times 60}{24 \times 60} = 47{,}000 \text{ cub. ft. per min. (approx.)}$$

Assuming also that there is a production of 155,000 cub. ft. of gas at N.T.P. per ton of pig, the volume of gas per hour is—

$$\frac{500 \times 155{,}000}{24} = 3{,}230{,}000 \text{ cub. ft.}$$

The gas is usually of a calorific value of approximately 100 B.Th.U. per cub. ft. For the purposes of calculation it will be assumed to be this value.

Stoves. Assuming the blast is heated from 60 to 1,200 ° F., the heat required to be given to the blast per minute is given by—

$$47{,}000\,\{(1200 - 60) \times \cdot 0189 + (1200^2 - 60^2) \times \cdot 0000009\}\text{ B.Th.U.}$$[*]

$$= 47{,}000 \times 22{\cdot}8 = 1{,}071{,}600 \text{ B.Th.U. per minute.}$$

Allowing for losses, the heat requirements are 70,000,000 B.Th.U. per hour. Assuming that the heat given up by each pound of brickwork in dropping 100° F. in the region of 1,200° F. is 28·9 B.Th.U., given by

$$\cdot 193(1250 - 1150) + \cdot 00004(1250^2 - 1150^2)$$[†]

the weight of checker work within 1 in. of the surface required in the stove, allowing for blast being on 1 hour is—

$$\frac{70{,}000{,}000}{28{\cdot}9 \times 2240} = 1080 \text{ tons.}$$

---

[*] Mean specific heat of air $C_p = \cdot 0189 + \cdot 0000009t^\circ$ F. per cub. ft.

[†] Mean specific heat of firebrick $= \cdot 193 + \cdot 00004t^\circ$ F. per lb.

If the stoves have an efficiency of 70 per cent, the volume of gas required per hour is 1,000,000 cub. ft., that is, with three stoves on gas and one on blast, each stove requires one-third million cub. ft. of gas per hour, neglecting the effect of sensible heat in the gas.

The area of the checker work exposed to gas and blast in each stove should be at least 60,000 sq. ft., the effective weight of the checkers being considered to be within 1 in. of the surface.

Engines. Assuming that the blast be delivered, if necessary, at a pressure of 18 lb. to the sq. in., the h.p. required at the furnace is—

$$\frac{47{,}000 \times 18 \times 144}{33{,}000} = 3700 \text{ theoretical h.p.}$$

Using reciprocating steam-engine and reciprocating blowers will require about 4,000 b.h.p., and similarly for a reciprocating blower driven by a gas engine. Owing to the lower efficiency of the turbo-blower itself, the b.h.p. required using a turbo-blower in conjunction with a steam turbine will be about 5,000.

Assuming that a gas engine uses, on an average load, 11,500 B.Th.U. per b.h.p. hour, i.e. at an efficiency of about 22 per cent, such an engine will consume about 460,000 cub. ft. of gas at N.T.P., whereas a turbo-blower consuming, at the boilers, say, 20,000 B.Th.U. per b.h.p. hour will require about 1 million cub. ft. of gas per hour at N.T.P.

Hoist. The power used in hoisting the burden is found as follows :—

The amount of material dealt with in 24 hours is—

Coke	500 tons
Ore	900 ,,
Limestone	250 ,,
Total	1,650 ,,

which is equivalent to 69 tons per hour.

Supposing the hoist can elevate 4 tons, taking 1 minute to hoist, the power necessary is, with a balanced skip,

$$\frac{4 \times 2240 \times 120}{33{,}000} \text{ h.p.}$$

where 120 ft. is the total height. This theoretical h.p. is 32·6, which could be dealt with by an electric motor of 40 h.p.

Pumps. Assuming that the furnace will need for all purposes 5,000,000 gals. per day, the water being pumped to a height of 60 ft. above the level of the tuyeres from a distance 25 ft. below, then, the weight of water per hour is—

$$\frac{5{,}000{,}000 \times 10}{24} = 2{,}080{,}000 \text{ lb.}$$

Work done by the pump

$$= \frac{2{,}080{,}000 \times 85}{33{,}000 \times 60} = 89 \text{ h.p. (theoretical).}$$

Assuming the pump efficiency to be 75 per cent and that the pump is actuated normally by electric motor, the latter will be of about 150 h.p.

Light and power may be estimated at about 300 h.p.

Thus, hoisting, pumping, lighting, and power generation may be summarized at about 500 h.p. and, if the generator be steam driven, the gas required at the boilers per hour would be : with dynamo efficiency, 90 per cent, and steam set, 8 per cent, 176,000 cub. ft.

Summary.

	Cub. ft. per hour	
Turbo-blower	1,000,000	= 30·9 per cent
Sundry	176,000	= 5·4 ,, ,,
Stoves	1,000,000	= 30·9 ,, ,,
Losses	162,000	= 5·0 ,, ,,
	2,338,000	

leaving about 890,000 cub. ft. per hour or 27 per cent for other purposes, or about 44 per cent when using gas engine blowers.

The gas distribution on a highly efficient plant is somewhat as follows—

Blowing	14–15 per cent
Stoves	25–30 ,, ,,
Pumps, hoist, and loss	5–10 ,, ,,
Total	44–55 ,, ,,

Thus, it will be seen that about 50 per cent of the gas evolved from the furnace is available for power production, or as gas fuel in metallurgical furnaces.

COKE AND ORE BINS

Although the handling of raw materials varies at different plants, each problem needing separate treatment, it is usual to supply bins near the furnace in order that the mixture comprising the furnace burden may be easily made and then rendered accessible to the lift, skip, or bucket charger. Owing to the quantities of materials dealt with in modern practice, the old wooden bunkers have given place to large steel bins supplied with suitable railroad discharge points above, and chutes at the bottom. Where mechanical charging is adopted, these chutes are designed to discharge into scale cars feeding the skip or directly into the bucket in bucket charging.

The shape of bins and the positions of the chutes are matters of great importance. For example, the latter should not be far apart, for ore, etc., will build up between them, rendering much of the volume of the bin useless. Then again, the treatment of sticky ore is different from that of rapidly flowing materials. The best form of gate for the chute is also a matter of great interest. It is clear that in order to stop the flow of ores from a chute, the gate, which usually is in the form of a quadrant, should rise above that point in the stream of material, such that its angle of repose is passed. In other words, owing to the retarding action of the gate, supplemented if necessary by a guillotine slide, the ore stream banks up. These points are of great practical importance, otherwise the scale cars are likely to be inundated with a flow from the chute which cannot be stopped until the bin is emptied.

It is claimed that a bin parabolic in section is theoretically correct as deforming stresses are eliminated. The bin capacity of the furnace mentioned in this article will be per 24 hours—

		Cub. ft.
Ores at 18 cub. ft. per ton	. 900 × 18	=16,200
Limestone at 22 cub. ft. per ton	250 × 22	= 5,500
Coke at 75 cub. ft. per ton	. 500 × 75	=37,500
		59,200

It is usual to allow for three or four days' supply of ore in case of breakdowns or adverse climatic conditions, and similarly, in the case of coke, where coke ovens are not attached to the furnace plant.

Stock Indicators

With hand-charged furnaces, the chargers measure the depth of the burden below the furnace top by means of metal lines provided with iron dippers. These are let down through small holes in the top, situated usually at four diagonal points through, or near the edge of, the hopper. When not in use the holes are closed with iron plugs.

One of the greatest difficulties in connection with the charging of a hand-charged furnace was to maintain the contents of the furnace at the proper level, non-observance of which rule causing serious derangement in the working of the furnace.

With mechanical charging, the stock level is gauged by means of a depth rod suspended by a rope passing over a pulley, and wound on a drum at the bottom of the furnace or in the hoist house.

In connection with similar apparatus, recorders have been devised which make suitable permanent records on circular or strip charts. Johnson's automatic stock recorder makes a continuous record as the rod rests normally on the burden, being drawn up automatically when the bell is lowered.

Pyrometry

Recording pyrometers are now installed for measuring the temperatures of blast and escaping gases.

An old form of pyrometer is that due to Uehling which, in principle, depends upon the difference in volume of heated air which is subsequently cooled to a fixed temperature. Electrical pyrometers, such as Callendar's resistance, Cambridge Scientific Instrument Company's base metal thermo-couple, and those made by the Bristol Recorder Company are now extensively used.

FITTING AND ERECTING

By Joseph G. Horner, A.M.I.Mech.E.

LESSON X

CRANE WORK—(*contd.*)

Overhead Travelling Cranes. The methods by which these are erected differ in some details from those of the smaller fixed, and travelling cranes. The design is that of girders bridging a shop, or yard, having wheels running on rails, on beams or gantries, and carrying the hoisting mechanism—the crab, or the jenny. An allied design is that of a tall framework or travelling gantry, which receives a crab, or a crane running on rails along its top girders, the whole structure moving on rails on the ground in a yard. There are many other cranes of large dimensions, in which horizontal girders carrying hoisting mechanism are the leading features—used about harbour construction works, docks, harbour service, coaling stations—the long-armed cantilever cranes, bridge cranes, titans, goliaths, differing in details, but having from the standpoint of the erector one feature in common, that of dimensions so large that levellings and alignments of related parts have to be effected by methods that differ in several respects from those which are suitable for the small cranes.

Taking the traveller as being generally typical of the long-armed cranes, it resembles these in the fact that it is built up with steel plating, angles and tees to resist bending under the

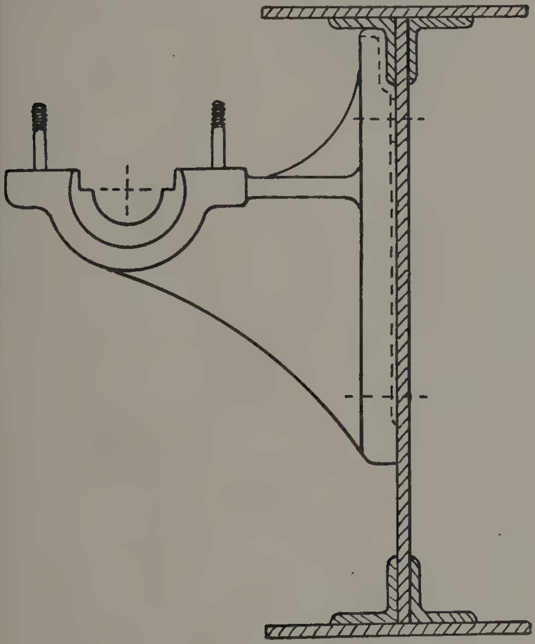
Fig. 62. Bracket to be Aligned on Girder

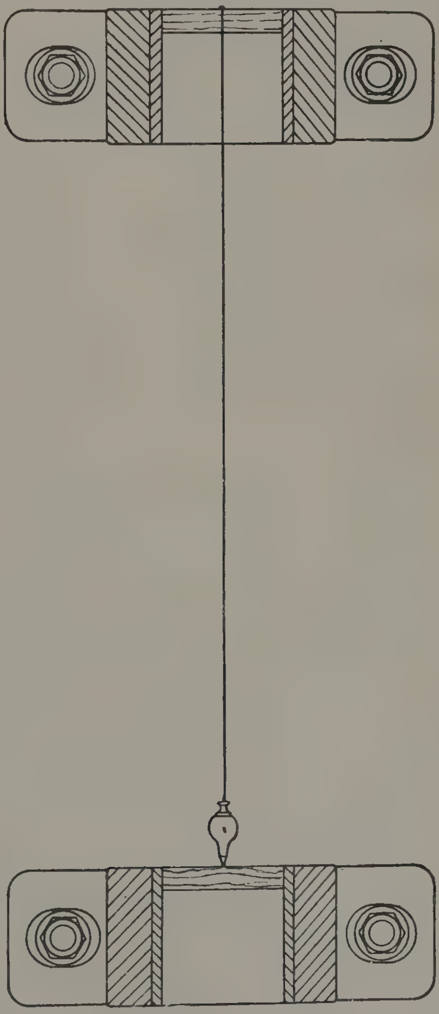
Fig. 63. Plumbing Bearing

imposition of load, and lateral flexure when travelling at high speeds. To these girders the erector has to attach end carriages with their travelling wheels, brackets to carry shafts, or cages for electrical equipments. Hand, steam, and electric power are all employed, and each effects material modifications in the character of the fittings, and the same may be said of the gantry and long-armed cranes generally.

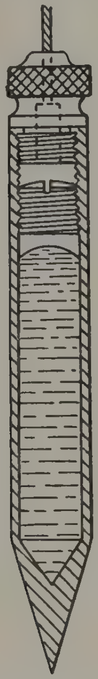
Fig. 64. Weighted Plumb Bob

Girders are connected at the ends only because the hoisting tackle has to descend between them. They are united with the end carriages, which are the first stages in the work of the fitter. These are generally plated, though in small cranes they are of cast iron. They are prepared complete before being riveted or bolted up to the undersides of the girders. They carry the bearings for the axles of the double-flanged travelling wheels, and for the spur wheels that drive. With these attached, the carriages are turned over, either on timber blocking, levelled, or mounted on rails over a pit, where the crane can be tested after the crab has been mounted.

Before commencing to fit any portions, it is well to test the girders for vertical and lateral truth. A level is hardly practicable unless it is laid on a long straight-edge, shifted into different longitudinal positions, and the level turned end for end. And a straining line will sag. A better method is to use a theodolite. Lateral truth is checked with a wire straining line pulled taut alongside the girder, which will show the slightest departure from linear truth.

When brackets have to be attached to the sides of girders to carry shafts, they are tooled first, the feet being planed or milled, and the bearings bored. The fitter then has to work by these. Corrections will certainly be required, their degree depending on the accuracy with which the work of the platers has been done. The plates may not have been levelled truly, flanges and angles may be slightly atwist, and castings like Fig. 62 will require adjustments, for which chipping strips are provided. A straining line pulled between the end bearings of a row indicates the amount of lateral inaccuracy. A length of shaft, or a parallel straight-edge checked with a spirit-level will show vertical relations. Or, when girders are of great length, a theodolite, sighting to the centres or the bottoms of the bearings affords an excellent test for making vertical corrections.

In the gantry and long-armed cranes, bearings for vertical shafts have to be set. The upper bearing or the lower one being fixed, the other is plumbed from it (Fig. 63). The bearings

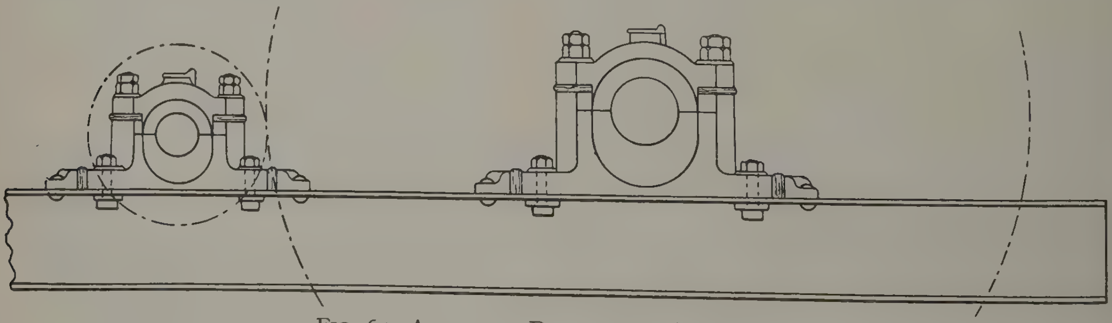
Fig. 65. Adjusting Bearings to Suit Gears

are bridged and the cord goes through the centre of one, and the point of the bob must touch the centre of the other. Plumbing is tedious because the bob is so long in coming to rest, especially out of doors where it is swayed by wind. It is steadied first by the hand until movement nearly subsides. To lessen the swaying, bobs should be heavy, of brass. The usual form is that shown in Fig. 63. Another is seen in Fig. 64, where it is hollow, made of steel and filled with mercury. The longer the line, the more necessary is a heavy bob.

When bearings have to be adjusted laterally, it is usual to have oblong bolt holes which permit of as much movement as they are likely to require. Adjustments of another kind are made in bearings where the centres of large gear wheels with cast teeth are liable to vary. To permit of accommodation, and to relieve some of the pressure on bolts, the bearings are carried in sole plates, planed to receive the feet, and having end abutment pieces against which adjustments for centres are made with packing strips of wood. Or separate abutments may be riveted to the steel girder or joist (Fig. 65), with the edges for the packings square as shown, or bevelled.

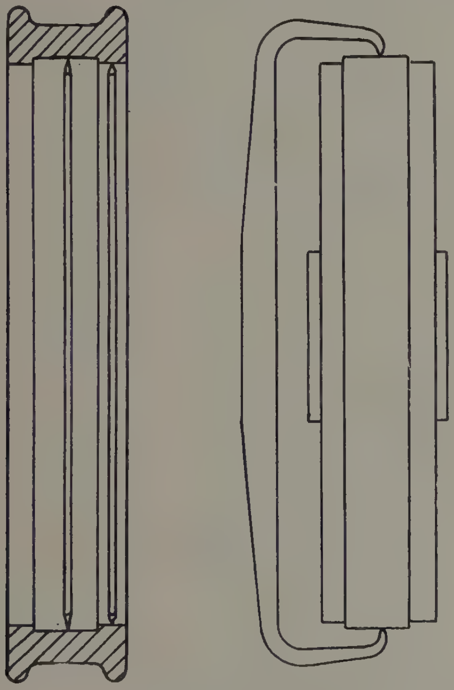
FIG. 66. FITTING TYRE TO WHEEL CENTRE

All massive wheels are tyred with steel on iron centres, or for the heaviest loads which will total to many tons on a wheel, on centres of cast steel. The boring and turning are done to gauges, Fig. 66, internal and external, the tyres for a 2 ft. 6 in. wheel being bored about $\frac{1}{8}$ in. smaller than the centre is turned. When expanded to a low red heat and shrunk on it holds without any other security.

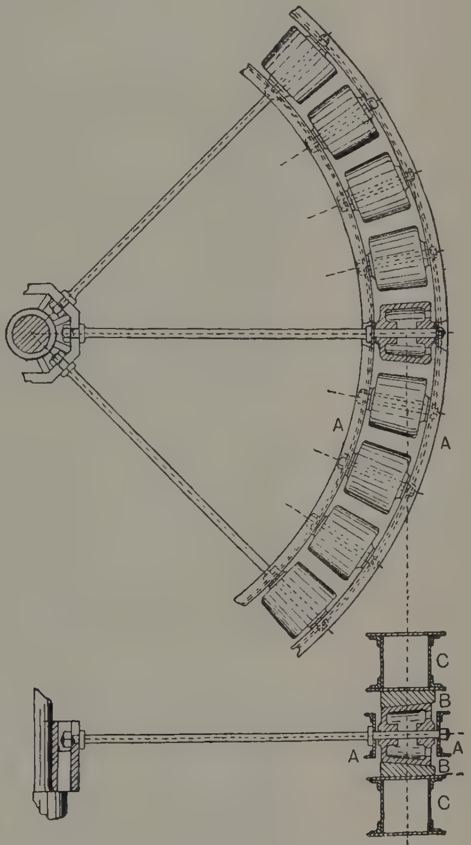

FIG. 67. ROLLER PATH FOR HEAVY CRANES

The Slewing of Massive Cranes. In the small portable cranes, the superstructure is revolved with rollers that make frictional contact with the post or with a path on the truck. In the large cranes it rotates on a ring of live rollers on the base, or on the top of the mast in the giant and titan cranes. The rollers are coned and the path similarly, the apex of the cone being in the centre of revolution. Diameters may range from 20 to 30 ft. The rollers are confined between top and bottom paths, and they run each on its own pin, by which they are kept apart. The rotation is effected with a separate ring of teeth, and the rollers sustain the load of all the upper works with the minimum of friction. A typical fitting is shown in Fig. 67. Two rings *A*, *A*, stiffened with angles, receive the pins on which the rollers run. The circular form of the rings is maintained by making the pins at intervals prolonga-tions of rods that are secured to a centra casting. The conical roller paths *B*, *B*, are fastened to circular box girders *C*, *C*, one of which is riveted to the base of the crane, the other to the underside of the revolving superstructure.

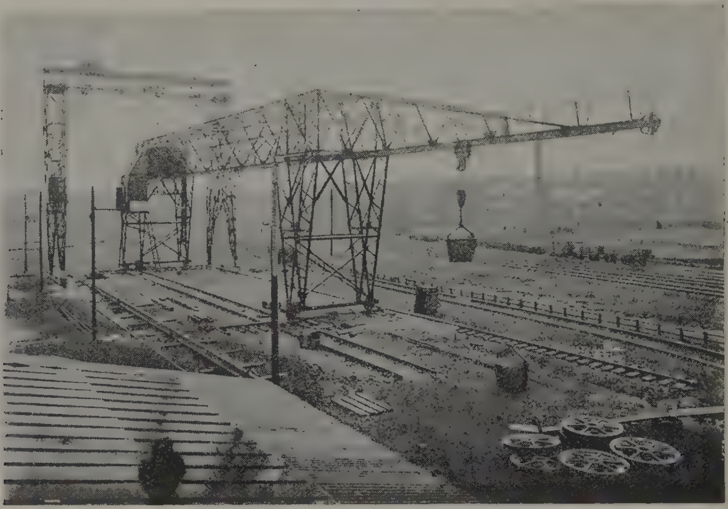

Fig. 68. Erecting Long-armed Crane
In the Yard of Messrs. Arrol & Co., Ltd.

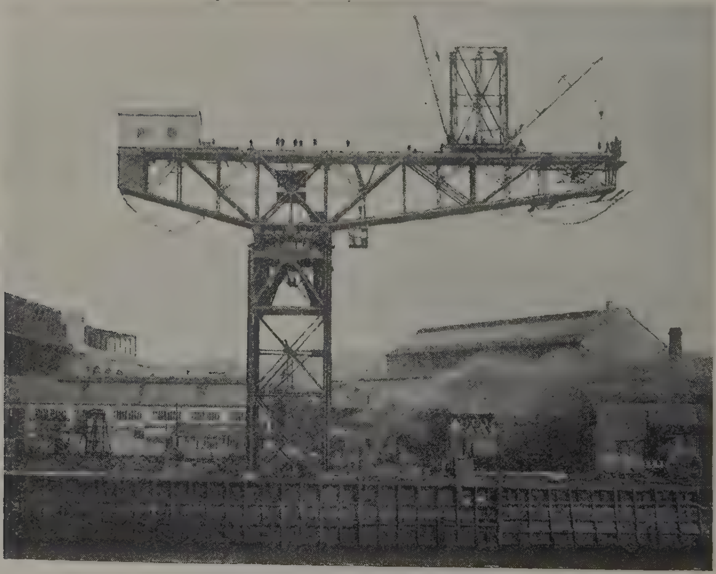

Fig. 69. Erecting Giant Crane
In the Yard of Messrs. Arrol & Co., Ltd.

Crabs and Jennies. These have protean forms. They are prepared complete apart from the traveller or long-armed crane to which they belong. The fitting does not differ materially from that of the superstructure of cranes. Some of the gantries carry a complete portable crane. Overhead travellers usually have crabs, but sometimes the more simple jenny which is fitted to the long-armed cranes, the latter arrangement being most desirable because then the mass of the operating mechanism is located behind the centre of rotation to serve as a counterbalance to the jib and the load. The crab carries all the hoisting mechanism. Its cheeks are usually built up of plate and angle, with bearings of cast iron, bushed, for the shafts of gears and drum. These are fitted and checked similarly to those of the cranes. When electric power is used, all the motions are controlled by the attendant in a cage at one end of the traveller or the jib. Two, three, or four motors may be used, distinct for travelling, traversing, and hoisting, and often more than one for light and heavy loads with different speeds. This work, and the wiring are handed over to the electrician.

Erecting Structures in Place. The outdoor erector must be a man of resource. There is much difference between building up structures in the works where overhead travelling cranes and travelling gantry cranes are available, and where every piece is lifted directly into its proper place, and re-erecting it away from these aids. Usually men of special experience in such tasks, and not the men in the shops are entrusted with outdoor erections. All the massive elements are marked and dismantled, but to locate and secure them again is often done with simple tackle. If a derrick is available or an adjacent crane large enough, there is little trouble. But very frequently there is nothing more than a

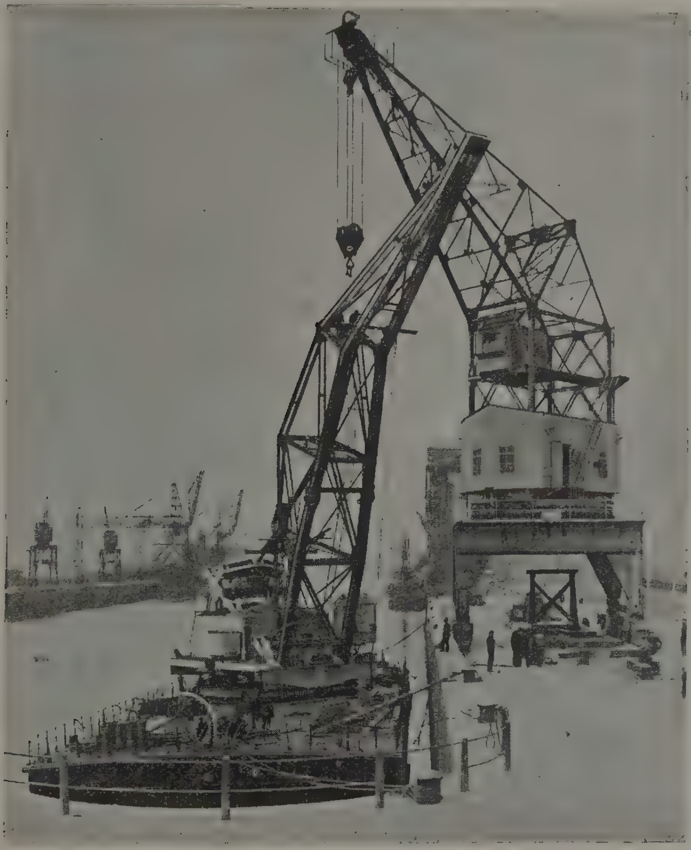

Fig. 70. Erecting Jib of Floating Crane in Dock
(Messrs. Arrol & Co., Ltd.)

pole, a single-sheer leg sustained with guys, and from which pulley blocks are suspended, or a guide pulley is hung from the top, and the hoisting power is obtained from a hand crab, whence a chain or rope passes over the pulley to a dependent hook or sling that is attached to the work.

EXAMPLES. An excellent grasp of what is involved in the erection of very massive cranes is to be obtained by the photographs in subsequent figures, for which I am indebted to the courtesy of Sir William Arrol & Co., Ltd., of Parkhead, Glasgow. Fig. 68 is a view taken in the works of the firm, showing an electric coaling transporter being erected with a 10 ton Goliath crane that travels the whole length of the yard. Fig. 69 illustrates the concluding stages of the construction of a 100 ton giant crane, often termed a hammer crane, in the works. Here a special stage with two derricks has been set up on the revolving arm to complete work that could not be covered with the gantry. Fig. 70 shows a 50 ton floating crane, having the jib placed in position with a 25 ton wharf crane, at Rothesay Dock, Glasgow. Fig. 71 is the same crane, having its jib lifted into position at its destination (Quebec) by means of a long pole. Fig. 72 illustrates a high shipyard tower crane, having a portion of the cantilever arm lifted into position by means of a framed steel derrick pole fitted with a short overhanging arm. The crane erector has to be a man of much resource.

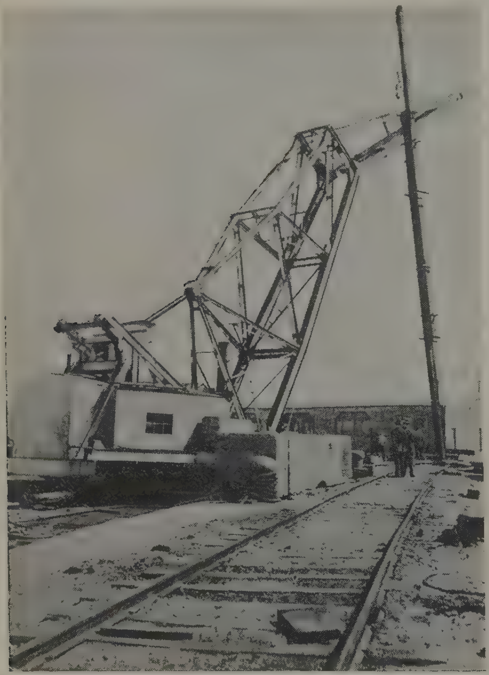

FIG. 71. ERECTING JIB OF FLOATING CRANE WITH A POLE

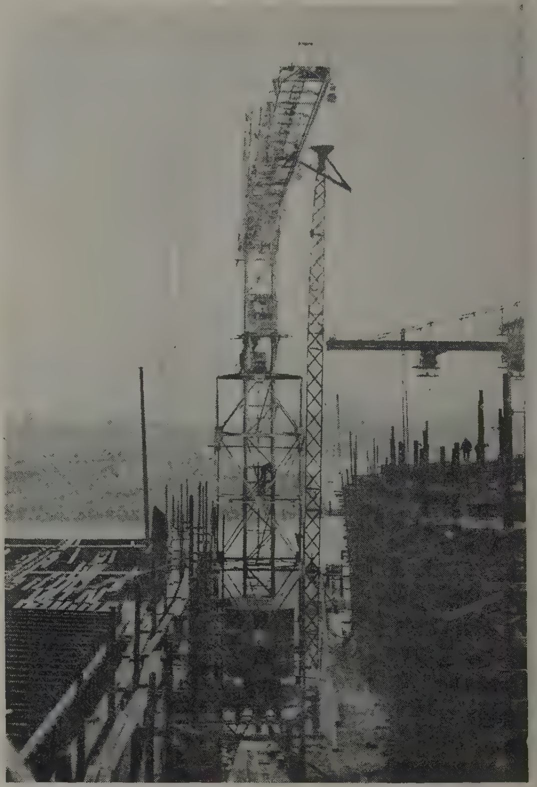

FIG. 72. ERECTING TOWER CRANE WITH A POLE

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc., (Vict.). M.I.A.E.

LESSON II

GENERAL REQUIREMENTS AND PRINCIPLES

In the previous lesson we have seen how, by a process of evolution and survival of the most popular, the automobile has arrived at its present outline. An automobile is divided into two parts: the chassis[1] or framework, which includes all the driving and controlling mechanism, and the axles and springs, which chassis supports the body or coachwork. Many early cars had the machinery added to a carriage, but in the coach trade there has always been a separation of coach body makers from the wheel, axle, and spring makers. This traditional classification has continued, especially as for commercial manufacturing the two classes of work are so dissimilar as to encourage the seemingly expensive method of production in separate factories. It permits greater elasticity of design in bodies without increasing the cost of those mechanical parts which involve much capital cost in production and, hence, require to be made in quantity for economical reasons. Many attempts have been made to produce "one-piece" vehicles, but the available body types are few and give the manufacturer little scope in catering for a changing fashion.

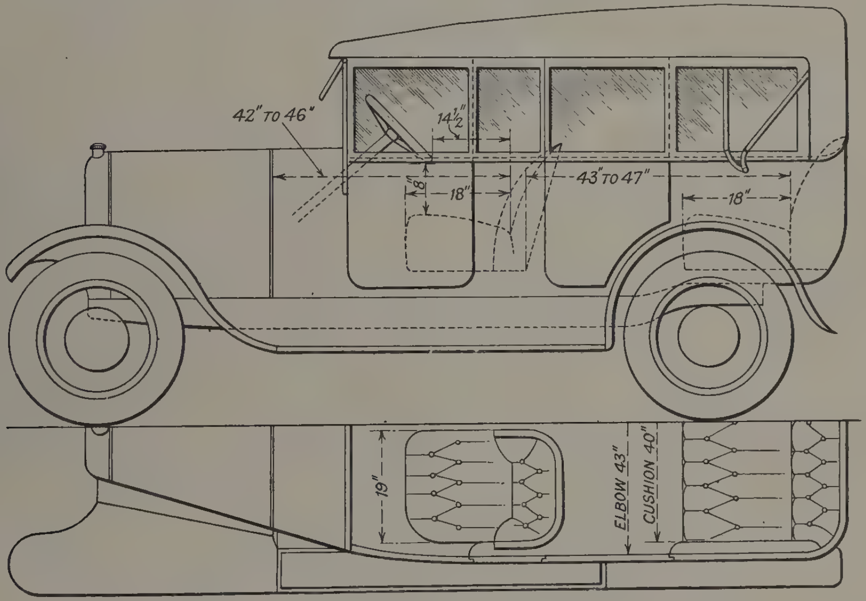

Fig. 4. Small Touring Car

Fig. 4 shows the general outline of a small car. The body dimensions given are about the smallest for the comfort of four people. Where only two are seated it is hardly worth shortening the frame as luggage is usually allowed for.

Fig. 5 shows the sizes of a large car seating two persons in front and three on the rear seat; although smaller dimensions can be used, those shown are general in good class body work. The difficulty in designing small cars is that their users are not smaller than the users of large cars. The more recumbent the seating position the lower the roof or hood and the longer the space required.

Fig. 6 shows the outline of a popular size of three ton lorry. The platform and frame height is limited by wheel size. Limits are also imposed by railway loading banks and wagon heights. There is a legal limit of 7 ft. 6 in. to vehicle widths in this class, but 6 ft. 6 in. is a usual width. Manufacturers object to long platforms as they are prone to overloading. With some goods tiering is not desirable, and long platforms have to be used for light loads. With vehicles over 25 ft. long, work is difficult at docks and railway yards unless a very large wheel lock is arranged for.

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[1] This French word strictly means a frame.

Fig. 7 shows a single deck omnibus. In this case there are legal requirements to consider. Seats are 16 in. long per person, and 2 ft. 2 in. knee room at seat level. Overall width is usually 7 ft. 2 in., although 7 ft. 6 in. is permitted outside London. The extra width is not often used however. Double deck omnibuses are at present undergoing many changes, due to the covering of the top deck, the introduction of pneumatic tyres and six wheels.

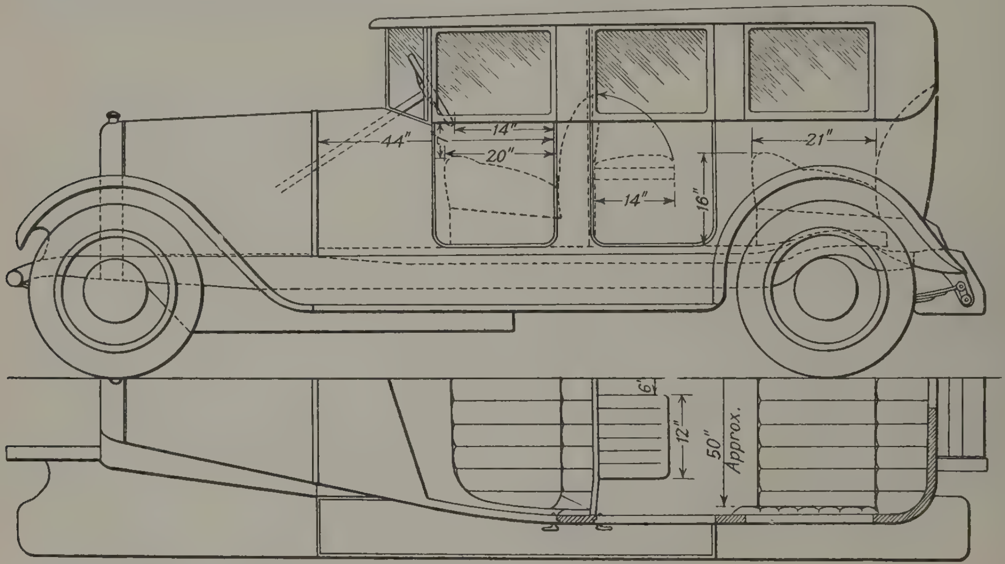

FIG. 5 APPROXIMATE DIMENSIONS FOR LARGE CAR OF WHEELBASE 12 FT., TRACK 4 FT. 9 IN., HAVING FOUR DOORS AND TWO DROP SEATS

THE PETROL ENGINE

Before proceeding to study the engine design it is necessary for the student to understand thoroughly the principle of its operation and the effect of change of design on performance.

The modern automobile is propelled by an internal combustion engine with at least four cylinders, six, eight, and twelve are also in use.

All automobile engines work on the same system, although there are substantial differences in construction between one make and another.

Each unit of a multiple cylinder engine is a separate engine. They draw fuel from a common source and mostly exhaust into a common pipe, but each has its own functioning parts.

If the student is not familiar with automobile work he should use opportunities of seeing first hand the interior parts of automobile engines and transmissions. No amount of illustration can convey the same impression as the real article, and to-day no other power production plant is so generally accessible for study as that of automobiles.

A typical engine is shown diagrammatically in Fig. 8.

A truly ground cylinder *a* has working in it a piston *b* which is attached by a connecting rod *c* to a crankshaft *d*. The rod is hinged at both ends. As the crankshaft rotates in its bearings, the pin *e* describes a circle and the piston is reciprocated in the cylinder.

The distance travelled by the piston is called its *stroke*.

The cross-sectional area of the cylinder multiplied by the stroke is called the swept or stroke volume, or piston displacement, or incorrectly is called the cylinder capacity. This is usually measured in cubic centimetres or litres. The student should estimate the cylinder capacity of various sizes of engines from the usual formula for cylinders—

$\text{Vol.} = \cdot 7854 \times (\text{diam.})^2 \times \text{stroke}$. Measurement in millimetres.

The top of the cylinder is closed by a head containing inlet and outlet passages called ports, which are closed by valves as shown. The piston does not reach to the top of the cylinder, the space left is called the compression space and is from one-fifth to one-third of the stroke volume.

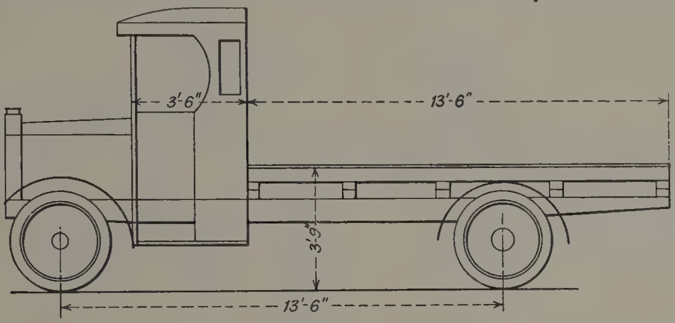

FIG. 6

The valves are kept closed by springs and are opened by "cams," which are irregular projections on a shaft driven at exactly half the speed of the crankshaft. Valves may be in many positions and operated in many ways, but the principle remains the same. The student should note in different designs the manner in which the opening is done.

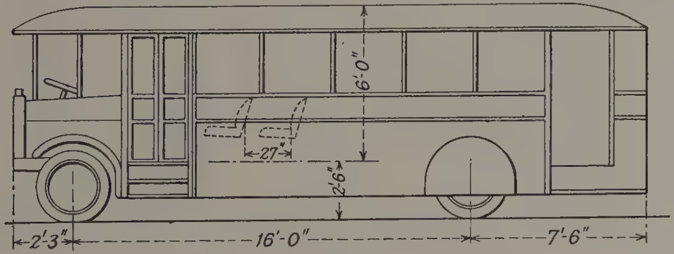

FIG. 7

In the compression chamber is introduced a wire electrically insulated from the cylinder metal and closely approaching a wire connected to the cylinder metal. Means are provided for passing at a determined time an electric spark across this gap. This insulated wire and gap is called a sparking plug.

On rotating the crankshaft by hand or by an electric motor, the valve *f* is opened when the piston commences to travel down, and remains open till it is at the bottom of the stroke. Air is drawn in along the induction pipe past a carburettor, where petroleum spirit (petrol) is sprayed into it, through the valve into the cylinder. At the bottom of the stroke the cylinder is full of a mixture of air and petrol vapour. The valve *f* closes and the piston travelling upward compresses the "mixture." In accordance with Boyle's Law the pressure rises, but owing to the mixture not being a perfect gas and the temperature not remaining constant, the law does not hold that absolute pressure varies inversely as the volume. The exact law will be studied later. At the top of the stroke the spark is passed at the plug points and the hydrocarbon burns in the air, producing steam and carbon-dioxide, which not only occupy at atmospheric pressure much greater volume than the oxygen and petrol vapour (the nitrogen of the air plays no part in the combustion) but the heat generated raises the temperature considerably, and these two causes produce greatly increased pressure in the combustion space. This forces the piston down and the pressure falls as the volume increases until the end of the stroke, when the valve *g* opens and the gases escape. The rising piston further ejects them and at the top of the stroke the valve *g* closes, *f* opens, and the cycle of operations goes on as before.

This cycle occupies four strokes, two up, two down, and is known as the four-stroke cycle. Engines have been made to operate on the two-stroke cycle and six-stroke cycle. The former is used for small motor-cycles and large stationary or marine engines, and the latter is not now in use. If we draw a diagram showing to scale the piston positions horizontally, and the pressures corresponding to those positions vertically, we get a graphic representation of what is going on in the cylinder.[1]

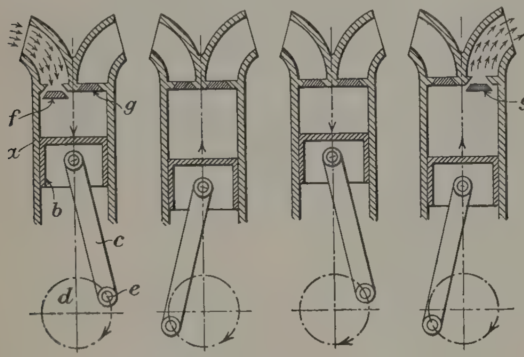

FIG. 8

This diagram is called an indicator card, and its study is essential to an understanding of the petrol engine. Work in its scientific meaning is proportional to force applied and to the distance moved by the point of application. It follows that as all vertical distances represent to some scale pressure per square inch they represent to some other scale *load on the piston*, and horizontal distances represent to some scale piston travel. So areas represent work done on or by the piston.

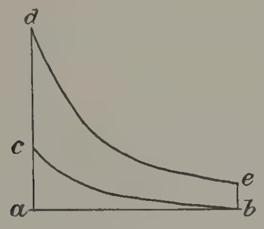

FIG. 9

Consider our diagram, Fig. 9. Owing to the slow speed no pressure is lost in inducing mixture to the cylinder ; the induction line is, therefore, at atmospheric pressure. (14·7 lb. per sq. in. absolute.) There is no leak or loss on compression or expansion, so the curves follow a law as shown. The exhaust stroke is slow and there is no back pressure, so that the exhaust follows the atmospheric line. The diagram, therefore, shows *adeb* as work done on the piston, and *acb* as work done by the piston. The balance *cdeb* is, therefore, net work. Any restriction of pipes will lower induction and raise exhaust lines, this is of opposite sign, and a diagram as Fig. 10 shows the plus and minus portions of a practical diagram. It is the designer's aim to keep the plus side as large as he can, and the minus side as small as he can, and our study of valve mechanisms will be undertaken in this light.

With a mental picture of the diagram the effect of speed, compression, ignition, or temperature changes can be studied.

If we had a diagram as Fig. 10, and drew a rectangle *jklm* whose area is equal to the net area of the diagram, i.e. positive part less negative part, then the height of this rectangle would be the average or mean height of the diagram. If this is translated into pounds per square inch to the chosen scale, then this becomes the mean effective pressure during a working cycle, as shown on the indicator card, and is known as indicated mean effective pressure (i.m.e.p.).

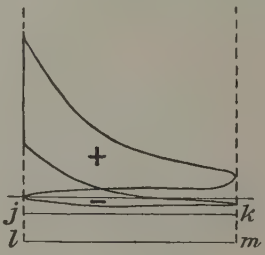

FIG. 10

If we have an engine with cylinder

D = inches diam. bore = A sq. in. area

L = inches stroke

N = revolutions per minute ; $\frac{N}{2}$ = cycles per minute

P = i.m.e.p. n = No. of cylinders.

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[1] In slow speed engines this can be automatically produced by a special pressure gauge marking on a card moved in unison with the piston, but the method is not applicable to fast running engines.

Then the mean pressure on the piston

$$= P \times A \text{ lb.}$$

$$\text{Work per stroke} = P \times A \times L \text{ lb. in.}$$

$$\text{Work per minute} = P \times A \times L \times \frac{N}{2} \times n \text{ lb. in.}$$

$$\text{Horse-power} = \frac{P \times A \times L \times \frac{N}{2} \times n}{396{,}000}$$

(One h.p. = 396,000 in. lb. per min.).

Horse-powers so calculated are called indicated horse-power, or i.h.p. For large slow running engines it is easy to obtain diagrams, but not easy to measure the power output. Indicated power is, therefore, used for the purpose of comparing such engines. In automobile engines the power output can be measured at the end of the crankshaft by friction brakes, by hydraulic brakes, or electrical brakes. Such power is called brake horse-power. It is less than indicated horse-power by losses in friction and driving auxiliaries. The relation on any engine $\dfrac{\text{brake horse-power}}{\text{indicated horse-power}}$ is called the mechanical efficiency of the engine. To increase power output, therefore, we can increase bore, stroke, i.m.e.p., revolutions per minute, or mechanical efficiency.

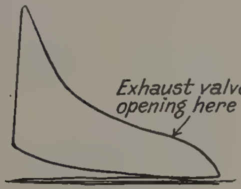


FIG. 11

These may react on each other. An increase in revolutions per minute may reduce i.m.e.p., and reduce mechanical efficiency. The power at which these reactions balance is the maximum horse-power output.

When running quickly, the statements made earlier as to the functions of valves require modification. Valves do not open and close instantaneously. Pipes and valves restrict the flow of gases, gases have inertia and energy. All these involve loss on the diagram as shown in Fig. 11. There is loss in induction, the line is below atmosphere, the inlet valve is kept open after the piston has started upward to allow gas to enter and to secure the highest charging efficiency. Owing to inefficiency of charging, the compression line is lowered. The fuel being less the expansion line is lowered proportionally. Late ignition causes losses.

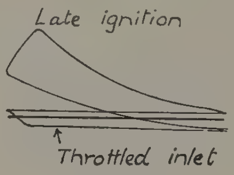


FIG. 12

If the exhaust valve is kept closed until the end of the stroke the pressure does not fall quickly enough and there is back pressure on the piston. As a compromise the valve is opened early, a small loss is suffered on the expansion stroke, and the larger loss on the exhaust stroke avoided. The diagram shows plainly where losses can occur. An extreme case of an engine running at a high speed is shown (Fig. 12). Little gas gets in and has to be "dragged in," it contains little fuel and pressure rise is low, the exhaust has to be pushed out. There arrives finally a speed where all the work done in the cylinder is used up in engine friction. This is the "limit of light running," and occurs at some speed above the speed for maximum output.

PATTERN-MAKING AND FOUNDRY WORK

By J. McLachlan and C. A. Otto

LESSON IX

CORES AND CORE MAKING

In its widest sense the term core is applied to any body of sand forming the internal part of a mould, whether formed as part of the mould or as a separate body of sand, but the term is more commonly applied to bodies of sand that are made separate from the mould and located in the mould after it has been prepared. Cores that are formed in the mould frequently have other terms to distinguish them from those made separately. Thus, part of a mould forming an internal shape, whether loose or not, is usually referred to as a " cod," while a loose part of a mould forming some external part of it, is generally referred to as a " false-core " or " drawback."

The use of cores in producing a mould may be a necessity or an expedient. They are necessary when it would be impossible to obtain an impression from a pattern made similar to the required casting, or, when it would be extremely difficult because of innumerable joints necessary in pattern and mould to produce such a mould. Even in many instances when a mould could be produced without cores, the pattern would be very frail or the sand forming internal parts of the mould would be likely to collapse before the rush and pressure of the metal. The successful venting of a mould would also present difficulties unless cores are used.

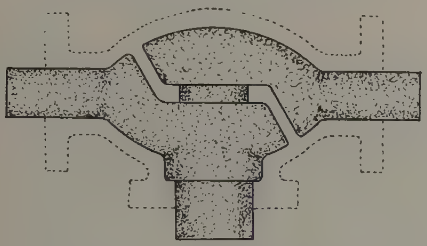
Used as an expedient cores may cheapen production, either by reducing the amount of pattern-work or in simplifying the making of a mould; they may be used to increase the number of distinct operations in order to speed up the rate of production by increasing the number of operations on a job. When the working space on the moulding floor is limited, cores may be used with advantage for practically the whole mould in order to reduce the time the floor space is in use for a particular job. Cores may also be used to facilitate modifications in similar types of castings that vary in length only, thus reducing the need for additional pattern construction.

Core Sands. It is necessary to exercise care and discretion in the selection and use of moulding sands, but it is even more important that suitable mixtures should be employed for making cores. Generally, a core is more nearly surrounded with metal at the time of casting, and the area through which gases can escape is more limited than with a mould; it is, therefore, important that the sand, of which it consists, should be open in texture. It is also an advantage that cores should crush during the time the casting is cooling to a normal temperature and so reduce contraction strains in a casting. When the casting has been made, the labour involved in removing the core should be reduced to a minimum. When the casting is light in section, up to about ½ in. thickness, the core should consist of a weak sand and be very porous.

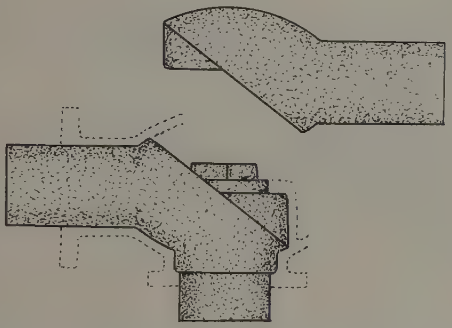
Naturally, bonded sands are generally used for medium and large sized cores when they are accessible for cleaning from the casting. The bonded strength can be varied according to the requirements of the work, but, whether strong or weak sands are necessary, they should have an open texture. For medium work, ordinary moulding sand may be used and sharp sand, may be added to weaken it or loam to strengthen it. To open the texture of a strong sand, coke, chopped straw, or sawdust, may be added.

There are several drawbacks to the use of clay bonded sands for cores. Clay reduces the permeability of cores, it sets hard after being baked and in the presence of the metal, in which condition it offers greater resistance to contraction and increases the difficulty of fettling the resulting casting. To obviate these difficulties, particularly for work of an intricate character, artificially bonded sands are now commonly used. These bonds invariably consist of carbonaceous materials which are more or less burnt from the core by the heat of the metal. At least partial disintegration takes place, and the core is more easily crushed by the contracting metal and is readily removed from the casting. The artificial bonds used are linseed oil, flour, core-gum, resin, molasses, peas-meal, besides many proprietary compounds sold for bonding foundry sands. Usually, sands very low in natural bond are used for preparing

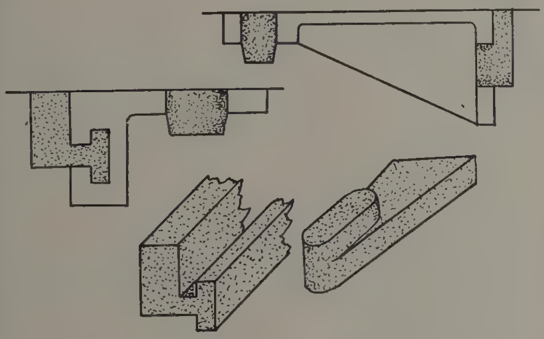
SOME EXAMPLES OF CASTINGS NECESSITATING CORES



Through Valve with Straight Division Plate, One Core Involved



Through Valve with Curved Division Plate, Two Cores Involved

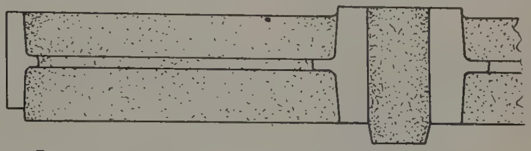


Tee-Slotted Angle Plate, Showing Part of Core for Slot

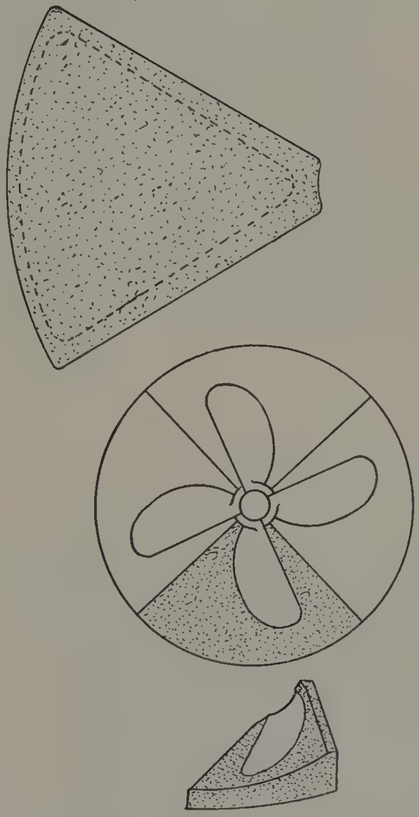
Simple Bracket Showing a Flange Core

Fig. 32

SOME EXAMPLES OF CASTINGS FOR WHICH CORES MAY BE USED AS AN EXPEDIENT



Large Wheel, Having Segmental Cores to Form Arms



Complete Mould for a Propeller Made with Section Cores

Half of Section Core Shown

Fig. 33

suitable mixtures, in many instances sea sand being used, particularly for intricate work. It is necessary to be more careful with the temperature of the oven in which the cores are baked when artificial bonds are used than for clay bonded cores.

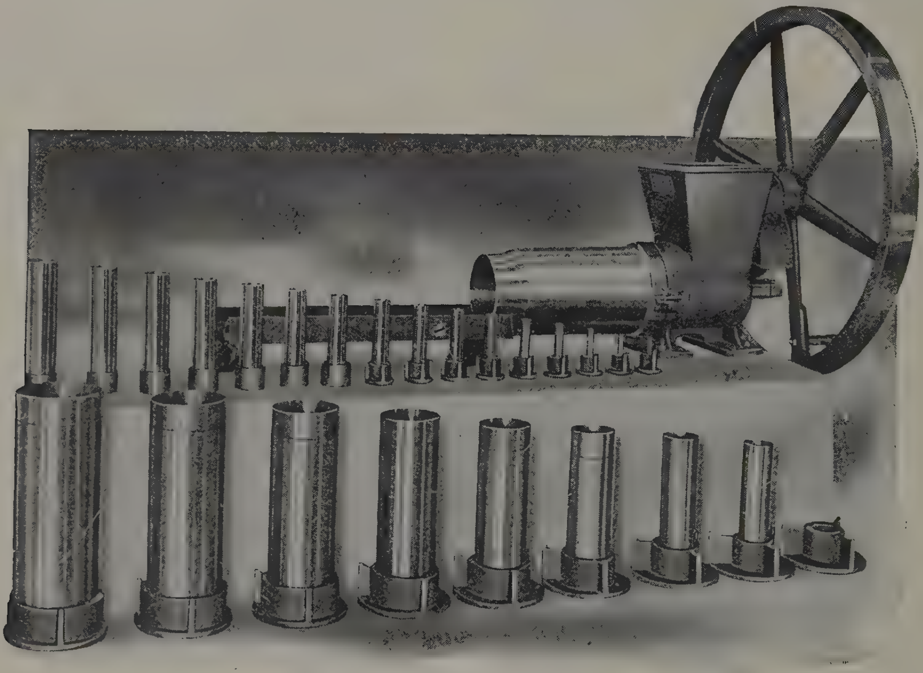


Fig. 34. Rotary Core Machine
(Messrs. J. W. Jackman, Manchester)

Core-making Machines. There are a number of machines of different types employed in the making of cores. The rotary or conveyor type is used solely for parallel cores, the sand being forced through a die having the sectional shape of the core. The sand is fed into the hopper on to a worm or conveyor, which, on being rotated, forces it through the die. Vents are formed in the cores as the sand passes the die. The plunger type of machine is not confined to the making of parallel cores: it can be used for shaped cores which can be stripped from their dies or core-boxes in a vertical direction. This necessitates loose pieces in the core-boxes and makes their construction more involved, unless the irregular shaped cores are of the simplest character. In this instance the die or core-box is fixed to a table, and over a plunger of the same sectional shape as the bottom of the die. Sand is rammed into the die by hand and then vented when the core is stripped by means of a plunger.

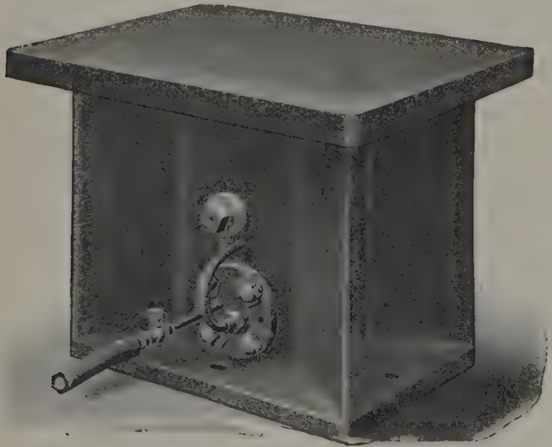


Fig. 35. Jarr-ramming Core Machine
(Messrs. J. W. Jackman, Manchester)

Another core-making machine, especially designed for pipe cores, prepares the cores on a spindle or core-barrel by means of a reciprocating steel bar as it revolves. The machine is operated mechanically, and as the spindle rotates the steel bar rams sand on it. The sand is fed to the machine by hand. Probably the most useful machine that can be used for general purposes is the core jarring machine. Any sized core-box may be used up to the capacity of the machine. The core-boxes are usually filled with the core sand by hand and then jarr-rammed with the machine, the stripping being done by hand. The hand or power press machine is also used for making cores, and, frequently some turnover device is used in conjunction with it to facilitate their production.

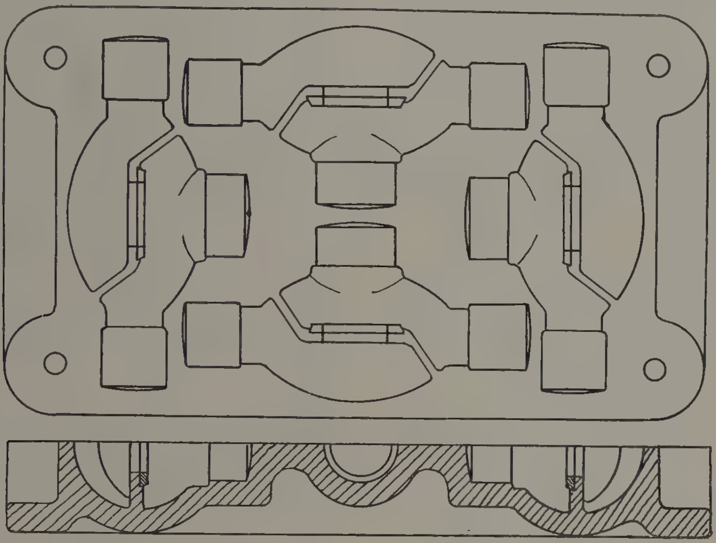


FIG. 36. CAST-IRON MULTIPLE CORE-BOX FOR SMALL VALVES

Cores for Repetition Work. When quantities of similar cores are required for repetition work, such as small valves, pipe fittings, etc., a number of cores are usually prepared at one operation from a multiple core-box, which carries the impressions for many half cores, the number depending upon their size. These multiple core-boxes may be of wood or some composition, but, frequently they are of metal. They may be prepared in halves with suitable guide pins for use on a press machine, or one-half may be used on a jar-ramming machine to produce half cores, which are afterwards pasted together. When an oil-bonded sand is used, the cores need to rest flat on a plate or be supported until they are baked. It is, therefore, convenient to prepare them in halves, venting before they are stripped so that they can rest flat on a plate, but, in order to save time in pasting, the complete cores may be made and supported on shaped plates for baking. Although oil-sand cores are very weak when green, they are very strong and easily handled after baking, and as long as the cores are sufficiently small to be handled without the aid of cranes, rodding or the use of grids to reinforce them is almost eliminated. Projecting parts of comparatively small sectional area may be wired or brads may be used, and these are generally sufficient to prevent breakages. Generally, when the cores

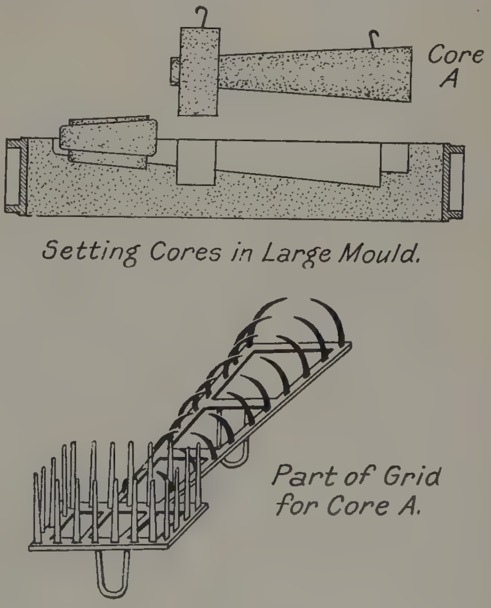


FIG. 37

are large and not capable of being handled, it is customary to make use of naturally bonded mixtures of sand, not because these are necessarily better for such cores, but because they are relatively cheaper than artificially bonded core-mixtures, and the economical production of castings must not be overlooked.

Core Grids. With naturally bonded sand cores in particular, some form of support is necessary so that they will have the requisite strength to facilitate handling or to be lifted by a crane. Rodding may suffice, or grids may be necessary according to the size and shape of the core. Whatever the form of support given to the core, whether of cast or wrought metal, it is important that consideration be given to the ease by which it can be removed from the finished casting. When the outlet from the casting is small, a wrought-iron support in the core can be pulled out, whereas a cast grid generally needs to be broken. Any support used in a core should offer as little resistance as possible to the core crushing under the strain of the metal when it is contracting. Wrought iron is especially useful when strains of this kind are likely to be troublesome as it gives to the strain, whereas a cast grid is rigid.

Venting Cores. Almost invariably, cores must be vented in some way in order to direct the gases generated at the time of casting. Air and gases will escape where least resistance is offered, and the core maker should see to it that the least resistance is likely to be encountered through that part of the core that will have least effect on the resulting casting. The ordinary vent wire is frequently used for making vents in small cores, or wax vents may be used. Combustible materials like hay or straw may be used as they are charred on baking and leave an opening in the core. For larger work the centre of the core may be filled with ashes forming a chamber, from which gases are exhausted through specially prepared exits.

STRUCTURAL STEELWORK

By W. Arnold Green, M.A., B.Sc., A.M.Inst.C.E., M.I.Struct.E.

LESSON II

STRUCTURAL STEEL—(*contd.*)

Plate Girder Stiffeners. In plate girders with a section such as that illustrated in Fig. 10, where the flanges are formed of pairs of angles riveted to the web plate at the top and bottom with or without plates or flats riveted to the horizontal outstanding legs of the angles, the most controversial part of the design is that of the web stiffeners. It might be thought that stiffeners should be placed in the diagonal direction in which the unstiffened web would tend to buckle, but from practical considerations this is never done. They are riveted to each side of the web at the ends and inner edges of the bearings, under concentrated loads and at centres determined by some rule. On the assumption that considerations of column action in the web should determine the spacing, and as experience has shown that a web whose depth does not exceed sixty times its thickness is satisfactory without stiffeners, rules have been devised, based on column[1] formulae, whereby the maximum allowable shear stress is permitted for the ratio $s/t = 60$, where s is the spacing of the stiffeners and t the web thickness. For greater values of s/t the allowable shear stress is lower.

In this country the rules given by B.E.S.A., No. 153, represent the best practice. Stiffeners, which are usually of angle or tee section, should be designed as struts of length three-quarters the girder depth to carry by a strut formula the total shear at the ends and two-thirds of the shear at intermediate points in the girder. If the clear depth of the web is more than sixty times the web thickness, stiffeners are required throughout at centres not exceeding the depth of the girder or 6 ft., whichever is the less.

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[1] Columns will be discussed in Lesson VI.

It is often specified that the ends of stiffeners should be ground to fit between the girder flanges at the top and bottom, and be attached to the web by enough rivets to transmit the load for which the stiffener is designed.

In wide plate girders the stiffeners are often bent by forging, so as to clear the flange angles as shown in Fig. 11, and thus considerably stiffen the flanges as well as the webs.

Long Spans and Deflection. As in short girders web shear stress may be the determining factor in the selection of a suitable section, so in long girders the allowable flange stress may cause undue deflection and, on that ground, a deeper section must be chosen or a lower flange stress used.

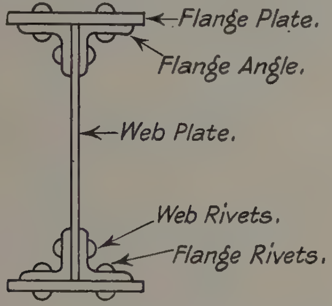


FIG. 10

The deflection of a girder may be expressed in the form

$$\triangle = \frac{W \cdot l^3}{C \cdot E \cdot I} \text{ or } \frac{f \cdot l^2}{K \cdot d \cdot E}$$

where

$\triangle$ = maximum deflection in inches due to the load.

W = total load in tons.

l = span in inches.

E = modulus of elasticity of steel, usually taken as 12,000 tons per sq. in., which is less than the actual to make a rough allowance for the deflection due to shear. For deep plate girders where the shear deflection is a greater proportion of the total 11,000 is sometimes the figure used.

I = moment of inertia of the section in (in.)^4, assumed constant throughout. Where not constant the adjustment is simple, though rather beyond the scope of these articles.

f = maximum flange stress in tons per sq. in.

d = overall depth of girder in inches. If the section is not symmetrical, twice the distance of the neutral axis from the stressed edge may be substituted for d.

C and K = constants depending on the nature of the loading = 76·8 and 4·8 respectively for uniformly distributed loads.

If $\triangle/l$ is to be limited to 1/360, i.e. 1 in. sag in a 30 ft. span, and $f = 8$ tons per sq. in. for a uniformly distributed load, $1 \div 360 = 8 \times l \div 4{\cdot}8 \times 12{,}000 \times d$, i.e. $l/d = 20$.

For a given load, span and maximum deflection, the girder may be selected by its moment of inertia which equals $W \cdot l^3 \div \triangle \cdot C \cdot E$.

In makers' handbooks, the safe loads for the longer spans are distinguished from those which produce a deflection less than a certain amount.

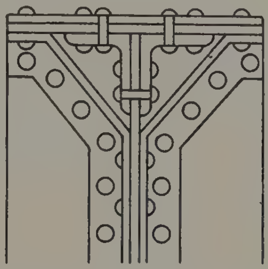


FIG. 11

Girder Depth. In selecting a girder, it should be borne in mind that a girder having a required section modulus is not necessarily the most suitable; as a deeper girder of less weight may have a greater section modulus than that required.

For plate girders a depth of about one-twelfth of the span is often adopted, and sometimes the girders are cambered slightly to prevent sagging when the load comes on to them.

The depth for the greatest economy varies with circumstances, but may be taken as between one-tenth and one-fourteenth of the span.

Flange Width. As the top flange of a freely supported girder is in compression throughout its length, there is a tendency for it to fail by side flexure if not properly stayed.

In ordinary building construction with a maximum working stress of 8 tons per sq. in., it is usual to limit the unsupported length to twenty times the flange width, reducing the working stress uniformly to, say, 5 tons per sq. in., when the ratio of unstayed length to flange breadth is 40. If the girder is concrete cased, it is reasonable to add the width of the casing on one side of the girder to width of steel to obtain the effective width.

B.E.S.A., No. 153, for plate girder bridges limits the unsupported length (l) to fifteen times the breadth (b), specifying on the gross area of the compression flange a maximum stress of $8(1 - 0{\cdot}0075\, l/b)$ if the flange has the outside edges stiffened with angles, and $8(1 - 0{\cdot}01\, l/b)$ if the edges are unstiffened.

Rivets. As has been mentioned in the note on quality, the steel used for rivets is more ductile than that from which structural sections are rolled. The diameter of the rivet near the head before being driven is usually specified, the size being given in eighths of an inch ($\frac{1}{2}$ in., $\frac{1}{4}$ in., $\frac{3}{4}$ in., $\frac{7}{8}$ in., etc.), and calculations are commonly based on these dimensions and not on that of the finished rivet, which after being driven red hot and upset through its entire length should completely fill the hole which is made with a diameter one-sixteenth of an inch greater than that of the rivet; B.E.S.A., No. 153, however, states that the rivet hole should be taken as the standard, but the former method makes an allowance for slight inaccuracies, unfortunately not unknown, whereby the holes in various members connected are not in complete alignment.

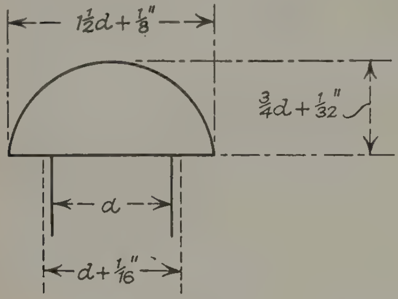


FIG. 12

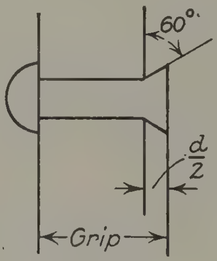


FIG. 13

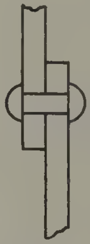


FIG. 14

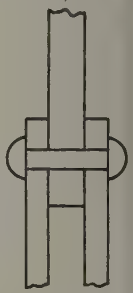


FIG. 15

Rivets commonly have cup heads of dimensions approximately as indicated in Fig. 12. Where it is necessary for a riveted surface to be flat, the heads are *countersunk*, and the shape of the head when chipped flush will be approximately as shown in Fig. 13.

Rivets may be used in *single shear* (see Fig. 14) or *double shear* (see Fig. 15). In practice, single rivets as indicated in the figure are rarely allowed.

Whatever working stress (B.E.S.A., No. 15 specifies 6 tons per sq. in., but 5 tons pe sq. in. is a quite common figure) is adopted f the shear stress in the rivet, twice this amoun is usually allowed as the bearing stress on th connected metal.

Thus, the area of a $\frac{7}{8}$ in. rivet is ·6013 sq. in and the value of the rivet in single shear a 5 tons per sq. in. is 3·01 tons, and in doubl shear 6·01 tons.

For bearing on a plate of thickness t in., th value is $\frac{7}{8}$ in. $\times\, t \times$ 10 tons, i.e. if $t =$ ·34 or ·6 the values in bearing are the same as those in single shear or double shear respectively.

For greater thicknesses, the shear value must be taken, and for less thicknesses th bearing values.

If used as indicated in Fig. 14, there will be a tendency for the plates to bend and put a tensil stress on the rivet. In addition there will be a bending tendency in the rivet itself. Owing t the shrinkage of a rivet as it cools there may b a considerable pressure between connecte plates and an indeterminable portion of th load will be transferred from one plate to anothe by means of friction.

It is thus seen that a riveted joint is not eas to analyse, and it is wise to adopt lower stresse if calculated as pure shear when it is known tha bending and tensile stresses also occur. Note on rivet spacing must be deferred to the nex lesson.

ENGINEERING SPECIFICATIONS

By Arthur P. Haslam, M.I.E.E.

LESSON VII

EXAMPLES OF ACTUAL SPECIFICATIONS

Steam Turbo-Alternator. In this concluding article it will be interesting to examine the specification issued by such a well-known firm as The English Electric Co., Ltd., for a large turbo-alternator set, and note some of the technical points to which they invite special attention.

Type. This is of the horizontal impulse type. The high pressure steam is confined to the stop valve, steam chest, steam nosing, and first nozzle plate.

Accessories Supplied. These are specified in detail.

Turbine Body. High pressure part to be cast steel, low pressure part of close-grained cast iron of a mixture specially suited for withstanding moderate temperatures, and designed so as to eliminate all distortion.

Provision for Expansion. Care is taken to anchor the turbine casing to the bedplate at one end, leaving the other free to slide and so adjust itself to changes in temperature. The diaphragms are of special quality cast iron, cast in halves and flexibly supported by keys to the turbine body. These keys are used in each half, so placed that expansion takes place away from the centre line, so that the diaphragm openings are always maintained co-axial with the shaft at all temperatures.

Rotor. The shaft is forged from high-quality steel and, during the process of machining, it is freed from all internal stresses by repeated annealing. All blading is from 3 to 5 per cent nickel steel, provided with a tang at the outer end of each blade by means of which the shrouding is riveted on. The shrouding is fitted in sections to allow for expansion.

Governor. The speed of the turbine will not vary more than—

Permanent variation between no-load and full-load and vice versa.	3 per cent.
Momentary variations between no-load and full-load and vice versa.	4 per cent.

This is provided for by a powerful totally enclosed centrifugal governor which controls the throttle-valve spindle by means of an oil relay gear. Hand adjustment provides for the speed of the turbine to be varied from 5 per cent below to 5 per cent above the normal speed. The governor is driven through worm gearing by the turbine shaft. The steam nozzles are divided up so as to form three batches. The first batch is controlled by the main throttle valve and allows the turbine to work up to about half load, when this load is exceeded the main throttle valve continues to rise and causes the first nozzle valve to open, which admits steam to the second batch of nozzles and permits the turbine to work up to full-load. Should overload be required the continued rising of the throttle valve opens the second nozzle valve and admits steam to the third batch of nozzles, and the turbine works under its maximum load conditions.

An entirely independent safety or emergency governor causes both the main stop valve and the main throttle valve to close, should the speed of the turbine exceed the normal by 10 per cent.

Lubricating. The oil-lubricating system with its oil pump for forcing the oil through all bearings, and its oil cooler, where the oil is cooled by flowing over pipes through which cold water circulates, is fully described.

Coupling. The coupling consists of two forged steel hubs bored and key seated, one being fitted to the turbine shaft and the other to the alternator shaft. These hubs have machined teeth on their periphery, and are surrounded by a muff in two pieces bolted together. The muff has two sets of internal teeth to correspond with those on the sleeves.

Alternator Stator. This is made of cast iron with flanged openings at top and bottom through either of which hot air can be expelled. The outside of this frame is covered with planished steel sheeting, and the inside is accurately machined to receive the stator laminations, dovetailed grooves being milled in the cast iron to receive corresponding dovetails forming part of the laminas. The laminations are insulated from the frame by moulded insulation of a specially hard nature placed in the dovetail slots.

Frame. The cast iron used for the frame has a minimum tensile strength of 8 tons per sq. in. The load required to break a 1 in. square test bar applied at the centre between supports 12 in. apart should not be less than 2,000 lb., with a deflection of not less than 0·1″.

Stator Laminations. These are of steel ·016 in. thick of high electrical resistance and low hysteresis loss. Each sheet is thinly insulated, additional insulations in the form of red paper being inserted at intervals. In addition to the internal slots for the windings they have holes specially spaced for axial ventilation.

The laminations are assembled under hydraulic pressure, the bolts being insulated from the laminations by insulation $\frac{1}{8}$ in. thick.

Rotor and Shaft. The rotor is of the cylindrical type, the shaft and rotor body being formed out of one solid steel forging having the following physical qualities—

Ultimate Strength = 40 tons per sq. in.
Yield Point = 20 ,, ,, ,,
Elongation = 22 per cent on shaft ends.
= 18 per cent on rotor body in tangential direction.
= 16 per cent on rotor body in radial direction.

The above tests to be made on a B.E.S.A. standard test piece. The special alloy steel end bells have the following physical properties—

Ultimate Strength = 58–62 tons per sq. in.
Yield Point = 50 ,, ,,
Elongation = 15 per cent on B.E.S.A. standard test piece.

The rotor fan blades are formed of high-grade steel about 0·128 in. thick pressed to shape while heated and riveted to the rings. The blade material has a minimum strength of 32 tons per sq. in. ultimate, 19 tons per sq. in. yield point, and 18 per cent elongation between 8 in. centres. The coils in both rotor and stator are of ample sectional area, subdivided to minimize eddy current loss, carefully placed in the slots, and thoroughly insulated after fixing to prevent any possibility of movement or of failure of insulation.

Exciter. This is of the overhung type shunt wound and provided with interpoles. The armature is pressed on an extension of the armature shaft, no coupling being employed. The armature laminations are built up on a cast-iron hub. The poles are of laminated steel punched with projections to support the field coils, and they are built up and riveted between steel plates. The commutator is of the "V" ring type, the segments being of hard drawn copper insulated from each other, and the steel "V" rings by means of micanite.

The commutator hub is in one piece with the armature hub so that any relative movement between the two is prevented. The armature and commutator can be drawn off the shaft as one complete unit.

In addition to the mechanical considerations which to a very large extent in these large generating units control the design, it will be noted that electrical considerations, such as low hysteresis loss in the rotor and stator laminations, and the size and sectional area of the individual electrical conductors are also stressed. On these depend, to a large extent, not only the normal output of the set, but, in addition, its power to withstand overload demands without getting too hot.

In smaller electrical dynamos and motors this temperature rise is very important, as it forms the touchstone by which differing types of machines can best be compared. The normal load of a machine is defined as the load it will develop when run for a given time without any part of its windings exceeding a temperature of a given number of degrees above the surrounding atmosphere.

It would be very interesting to continue an examination of commercial specifications for different engineering requirements, they would be found to be based upon the principles and considerations which have been briefly touched upon in these articles. The main points to remember are that for successful engineering work it is absolutely necessary to have means of securing material of uniform quality, and, having these, for the engineer to maintain an open mind as to new ways of using them to the best advantage.

(Conclusion)

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XVII

THE AGE OF HIGH GRADE STEELS

SIR HENRY BESSEMER (1813-1898)

In the year 1856, Bessemer read his now celebrated paper on "The Manufacture of Malleable Iron and Steel with Fuel" before the British Association. The essentials of the new process of steel manufacture put forward in the paper aroused a good deal of scientific interest; yet, at the same time, the proposed new process was received with very little favour by the steel trade itself.

Sir Henry Bessemer

Bessemer, however, was not to be discouraged by this latter fact. He set up for himself in the manufacture of steel, his first factory being situated at Sheffield. And it is now almost unnecessary to call the reader's attention to the degree of success which eventually overtook this undaunted metallurgical pioneer. Bessemer steels are world famous, and the Bessemer process, in a somewhat modified form, is the one process which is now in universal employment for the large scale production of high grade steels.

Briefly, the original Bessemer steel process consisted in blowing compressed air through molten pig-iron, which was contained in a vessel known as a "converter." The carbon contained in the metal, together with the other impurities, are oxidized away by the compressed air blast. Finally, after the passage of the air blast, the contents of the converters are run out into specially prepared clay moulds. As a rule, the Bessemer converters are worked in pairs, as shown in the illustration on this page. One of the converters receives the molten charges, whilst the other is moved into a vertical position ready for the compressed air, or "blast" treatment.

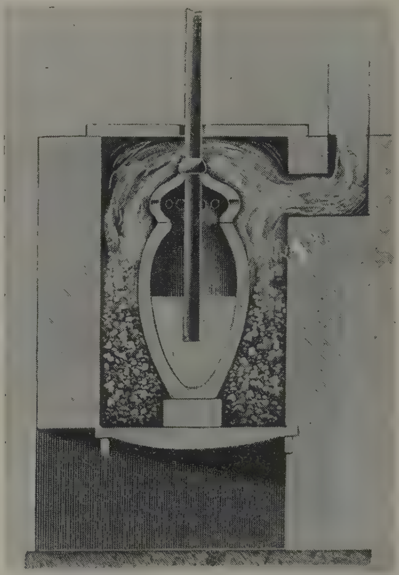
A Pair of Bessemer "Converters" in Action

The originator of the process, Henry Bessemer, was born in 1813, the son of a certain Anthony Bessemer, an engineer who had had active experience of steam plant erection in this country and in Holland. The young Bessemer possessed a mechanical and an inventive turn of mind, and quickly began to follow in, and, indeed, to improve upon, his father's footsteps.

As a young man, he devised several useful processes in the various industrial arts. He brought out improvements in the then fashionable electrotyping industry, by means of which plaster casts, medals, and similar articles were copied accurately in metal. Bessemer invented processes for the manufacture of improved types of alloys. He turned his attention to the subject of glass manufacture and sugar refining, in both of which processes he devised improvements upon the existing methods. The stamping of official, financial, and legal documents was a subject which also attracted Bessemer's attention, and he made such advances towards the attainment of an ideal "forgery proof" document that it was very largely on account of his work in this direction that he received the honour of knighthood.

A Victorian Cartoon of Bessemer

It would be impossible to enumerate within the space at disposal a tithe of the improvements and inventions which Bessemer brought out in one field of applied science or another. In all, his published patent specifications amount to nearly two hundred. Even after his retirement from business, Bessemer busied himself in various occupations and active hobbies, one of the chief of which was the subject of astronomy. He constructed several large telescopes, and was busy with the plans for further instruments of this type when death finally called him away from the scene of his labours on the 15th of March, 1898.



Section through Bessemer Crucible

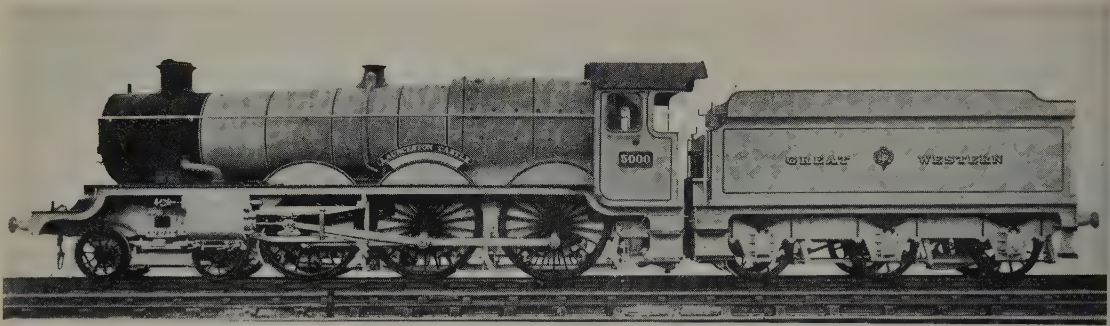


FIG. 3. 4–6–0 EXPRESS LOCOMOTIVE (Great Western Railway)

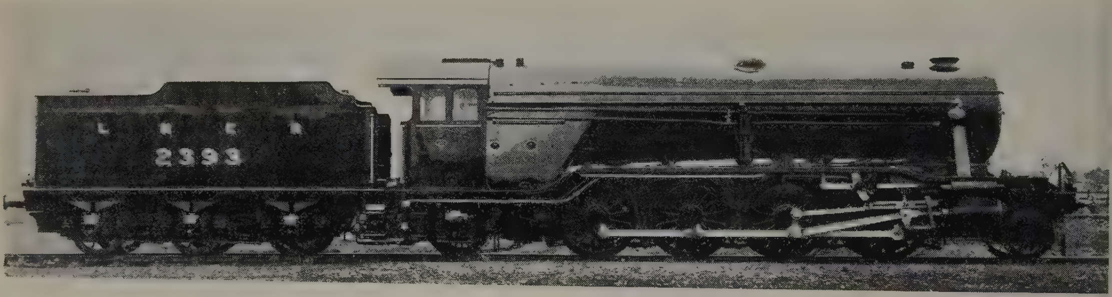


FIG. 4. 2–8–2 GOODS LOCOMOTIVE (London & North Eastern Railway)

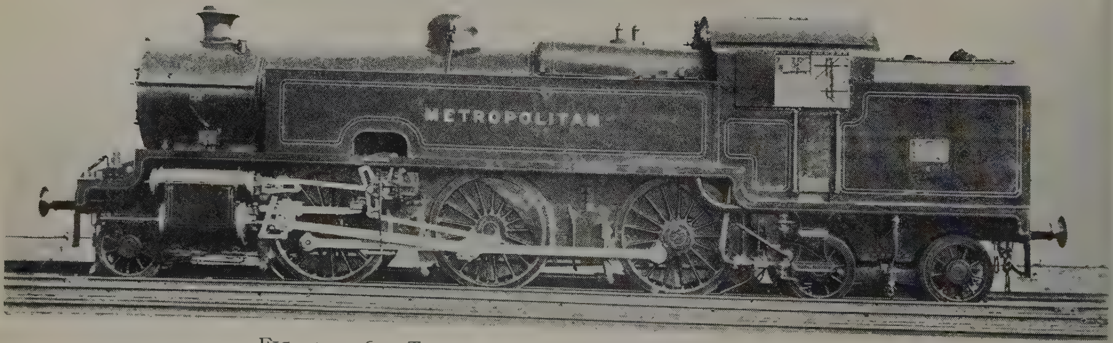


FIG. 5. 2–6–4 TANK LOCOMOTIVE (Metropolitan Railway)

OUTLINES OF LOCOMOTIVE ENGINEERING

By A. Morton Bell, O.B.E., M.I.Mech.E., M.I.Loco.E.

LESSON II

GENERAL DESCRIPTION

The modern locomotive may be described as a very closely built and compact, portable power plant, the designers of which are seriously hampered by restrictions in over-all size, etc., not applicable to marine, stationary, and other examples of power generating machinery. Further, locomotive engineering recognizes no standards of high economy, for the reason that the results obtained from locomotives vary so much according to the different conditions of load, speed, and weather. Isolated trials and experimental runs cannot be taken as more than representing approximately actual results in practice.

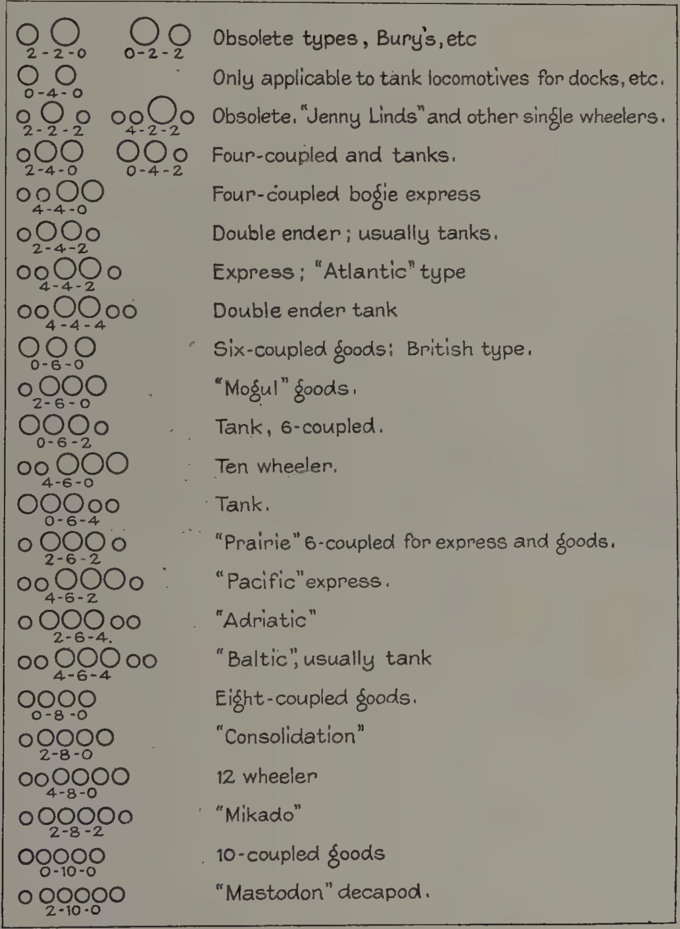


Fig. 2

It is customary to record the types of locomotives by enumerating their wheel notation in a group of three figures, the first unit representing the number of leading carrying wheels, the second the number of driving wheels, and the third the number of trailing carrying wheels. For example, the indication "2–2–2" signifies an engine with a single pair of carrying wheels in front, a pair of driving wheels, and a single pair of trailing carrying wheels. The indication "0–6–0" represents an engine with 6-coupled drivers, and no other carrying wheels, leading or trailing.

The different wheel arrangements more commonly found in practice are as the table given by Fig. 2.

Early types of locomotives included 4-wheelers, 2–2–0, 0–4–0, etc., but these have all but disappeared from railway service, the only examples of the latter type now running are to be found in tank locomotives used in industrial works, docks, etc., of which an example will be given.

The 2–2–2 and 4–2–2 types of single-wheel express engine, too, have become obsolete; in fact, modern traffic requirements have completely outclassed all types of single-driver engine, owing to their limited adhesion properties. Various types of 4-wheels coupled locomotives have since become general for passenger trains, but these are, in turn, now being rapidly replaced by engines having six, or more, driving wheels coupled.

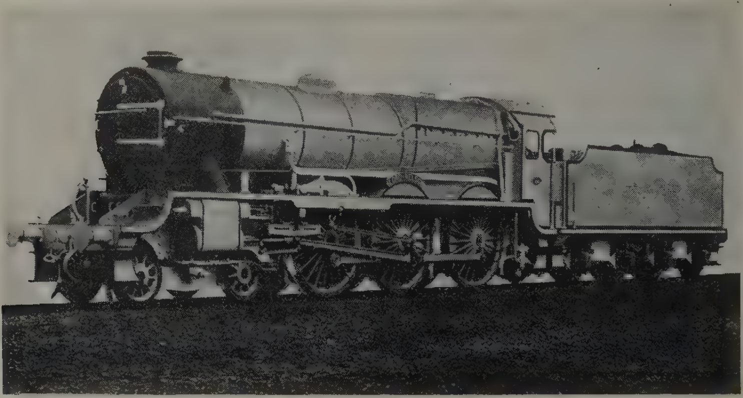


Fig. 6. 4–6–0 Express Locomotive (Southern Railway)

By way of illustrating typical examples of modern British express locomotives, a 4–6–0 of the Great Western Railway and a 4–6–0 of the Southern Railway have been chosen. These engines probably represent two of the most efficient locomotives which British designers have so far produced, their boilers and engine power being well proportioned to the adhesion. In the case of the Great Western Railway engine, the steam pressure of the boiler is 225 lb. per sq. in., its cylinders (4) are 16 in. in diameter by 26 in. stroke, and its driving wheels 6 ft. 8½ in. in diameter. The chief features to be noted are the tapering form of the boiler, to secure ample steam space; the position of the feed, to assist in securing good circulation of the water, as also the arrangement of the superheater; four cylinders, to provide sufficient power with uniform turning force to the cranks; and the very efficient valve gear, to ensure satisfactory distribution of steam must also be noted. The weight of the engine in running order is 79 tons, and that of the tender, with 6 tons of coal and 4,000 gal. of water, 46½ tons.

The Southern Railway engine carries a working pressure in the boiler of 220 lb. It has four cylinders 16½ in. diameter by 26 in. stroke, and driving wheels 6 ft. 7 in. diameter. In this case the boiler is parallel, and is provided with a dome, whilst the firebox is of the "Belpaire," or square, flat-topped type. The weight of this engine in running order is 83½ tons, and that of the tender, with 5 tons of coal and 5,000 gal. of water, is 56½ tons. The distribution of weight in this case and the limited load of just over 20 tons on each driving axle permits of this engine being used on any of the main lines of the railway it is built for.

Although custom has been followed in taking for typical illustration express passenger engines, there is another class of locomotive on which a railway probably depends more for revenue earning than any other, and that is the goods engine. In this country 6 and 8-coupled locomotives are general, and provide sufficient haulage capacity, but abroad the loads, in many cases, can only be handled with engines of much greater power. So long as British railways are restricted to the loads they at present cope with, it is difficult to see the necessity for anything much heavier or larger than the engines which are now employed, although, for special purposes, locomotives have been designed of increased power; these will be mentioned more in detail in a later section of these lessons.

The goods locomotive, Fig. 4, is one of the 2–8–2 type for the L. & N.E.R. with a superheater, and may be taken as an example of the latest design and construction. Similar remarks appertain to the boiler, superheater, etc., as those made when noting the passenger engines illustrated. The cylinders (3) are 20 in. diameter by 26 in. stroke, whilst the driving wheels of 5 ft. 2 in. diameter are sufficiently large to enable the engine to give a good account of itself if called upon to perform passenger service; the boiler carries a working pressure of 180 lb. per sq. in. The weight of the engine in running order is 100 tons, and the tender weighs 51·5 tons, with 4,700 gal. of water and 7 tons of coal. The engine is equipped with a "booster," a comparatively recent development in locomotive design, although forecast years ago by steam-driven tenders, etc. The modern arrangement consists of a small engine, having two cylinders 10 in. diameter by 12 in. stroke, which drive the trailing pair of carrying wheels of the engine through a system of gearing, whereby they can be thrown in and out of service, as required. By utilizing the additional tractive power of this booster, the pulling effort of the engine can be temporarily augmented to enable it to easily start heavy freight trains weighing approximately 1,690 tons, on grades of 1 in 200. Under normal conditions, the tractive effort of the engine shown is 38,500 lb., but, with the booster in operation, this is increased to 47,000 lb.

A third type of locomotive which must be mentioned in this brief enumeration of types is the "tank" engine, much used for suburban, local, and auxiliary services. Fig. 5 shows a 2–6–4 of the Metropolitan Railway (London), representing an engine of a more than ordinary useful character. Such engines are employed on fast passenger, as well as more bulky and slower freight trains. The boiler has a working pressure of 200 lb. per sq. in.; the cylinders (2) are 19 in. diameter by 28 in. stroke, whilst the driving wheels are 5 ft. 6 in. diameter. The weight in working order, with 2,000 gal. of water in the tanks and 4 tons of coal in the bunker, is 87 tons. The tractive force is calculated at 26,000 lb.

One other leading type of normal locomotive should be considered here—the 4-wheeled tank engine (0–4–0), already referred to as being

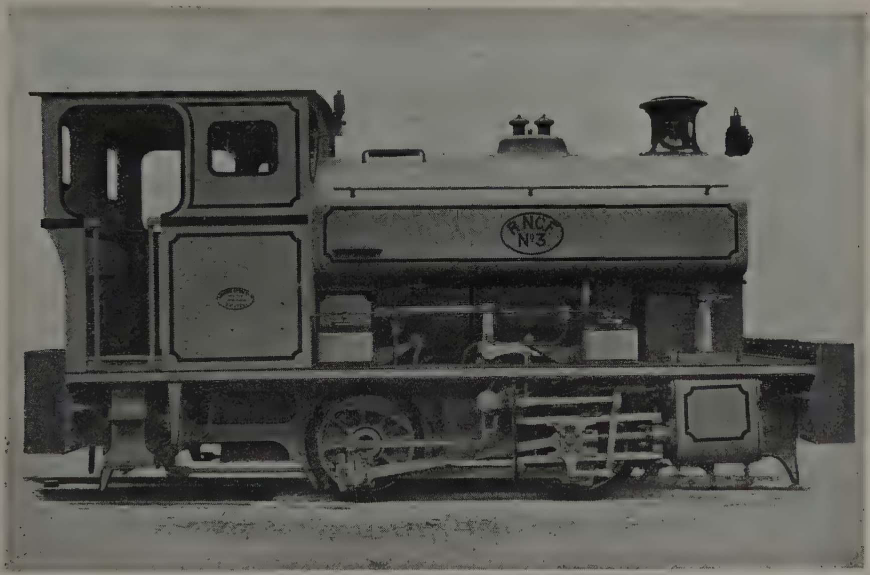


FIG. 7. 4-WHEELED TANK ENGINE

much used in factories, collieries, etc. The reproduced photograph shows one of a standard make, popular, due to its simplicity, straightforward arrangement of parts, etc. It has cylinders 13 in. diameter by 18 in. stroke, and driving wheels 3 ft. diameter. The boiler pressure is 180 lb. per sq. in. An engine of this description is capable of hauling a train of 800 tons on level sidings, and can negotiate loads of 180 tons on a grade of 1 in 50. The total weight in running order is 26 tons.

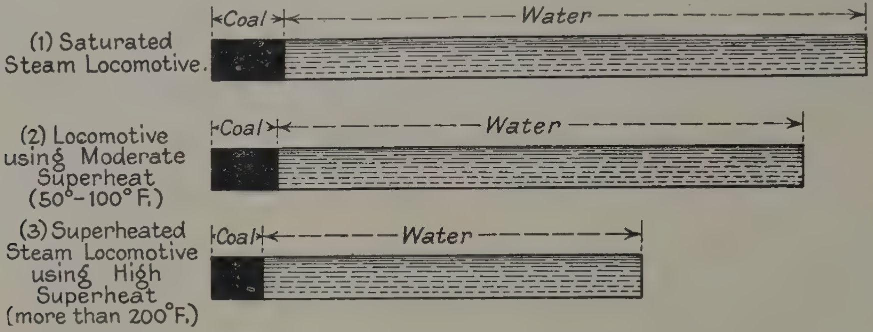
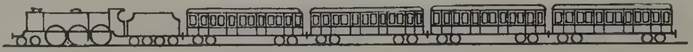
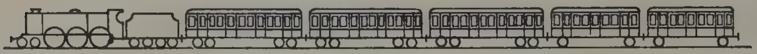

Coal
Water
(1) Saturated Steam Locomotive.
Coal
Water
(2) Locomotive using Moderate Superheat (50°-100° F.)
Coal
Water
(3) Superheated Steam Locomotive using High Superheat (more than 200°F.)

FIG. 8

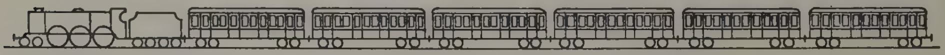
So far the locomotives represented and described have been of the "simple," or single expansion type, in contradistinction to "compound," or double expansion, but in each case, with the exception of the last, they have the modern refinement of superheating; it will not therefore be out of place to make a few remarks here on this particular development before treating on compounding.



(1) Saturated Steam Locomotive.



(2) Locomotive using Moderate Superheat (50°-100°F.)



(3) Superheated Steam Locomotive using High Superheat (more than 200°F.)

FIG. 8A

Although the advantage of drying and superheating the steam generated in the boiler, after removal from contact with the water, has been long appreciated, there were practical difficulties in dealing with steam of high temperature, especially in the lubrication of the surfaces of the valves, cylinders, etc., hence the delay in general adoption.

The chief advantages claimed for superheating may be summed up in the following—

1. Perfected nature of the steam after its generation in the boiler, and its increase in temperature; delivered to the engine more in the form of a gas.

2. No condensation in the engine cylinders, with consequent economy in fuel.

3. Less weight of steam required for given amount of work.

4. Increased tractive effort of the locomotive.

The above diagrams, Figs. 8 and 8A, will effectively illustrate these claims.

The utilization of the advantages mentioned have formed the problem confronting those who have been instrumental in perfecting the present superheating arrangements. The illustration, Fig. 9, shows in section the application of the superheater of the fire tube type to a locomotive boiler.

Compounding, which for a time was a very popular method of increasing the resultant efficiency of the steam generated in the boiler of a locomotive, provides a means whereby the range of expansion can be increased to secure more useful effect in the cylinders. Prior to the perfecting of superheating devices, compounding had many adherents, but as the benefits accruing were somewhat discounted by the necessity for employing a multiplicity of cylinders and parts, compound locomotives are no longer in the favour they were, although many interesting and successful applications exist.

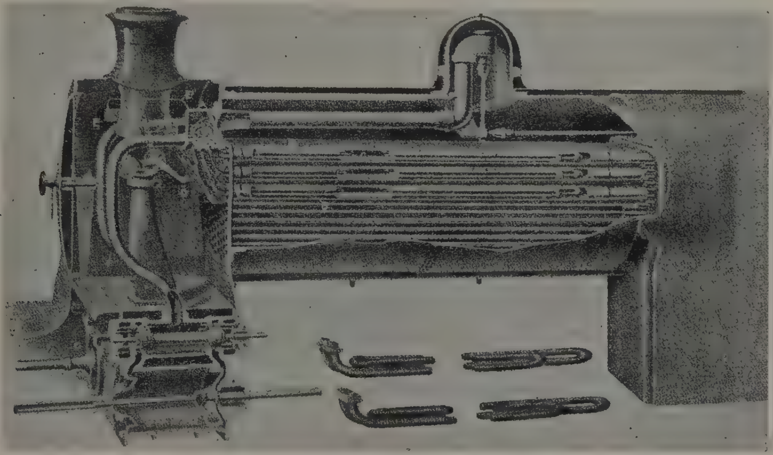


Fig. 9. Schmidt Fire-tube Superheater with Four-fold Elements

The advantages claimed are as follows—

1. Economical use of steam.

2. Range of expansion per cylinder reduced, and consequent decrease in range of temperature, cylinder condensation considerably lessened.

3. Total range of expansion increased.

4. Uniformity of work on pistons, and, in the case of three- or four-cylinder engines, more uniform rotation of axle.

A great variety of cylinder arrangements have been tried, the more common being as follows—

1. Two cylinders: one, the high pressure, about normal size; the other, low pressure, larger.

2. Three cylinders: one high pressure and two low pressure, all about the same diameter. Perhaps this is one of the most successful forms, and is still much favoured.

3. Three cylinders: two high pressure and one low pressure much larger.

4. Four cylinders: two high pressure and two low pressure.

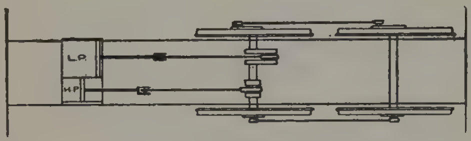
All arranged to drive one one axle, or to drive separate coupled axles. Further, the cylinders can be arranged tandem, superimposed, or carried on articulated framing.

The sketches in Fig. 10 outline their arrangements, and give some idea as to the variety which can be adopted for compound locomotives.

Locomotives built as two-cylinder compounds (one H.P., and one L.P. cylinder) had to be provided with some special gear for starting should the engine be standing on the dead centre of the one H.P. cylinder. Much ingenuity was displayed in the design, and several forms, comprising an intercepting valve to shut off communication for the time being between the H.P. and L.P. cylinders, whilst admitting boiler steam to the latter, at the same time as the ordinary regulator allowed steam to enter the H.P. cylinder, were adopted.

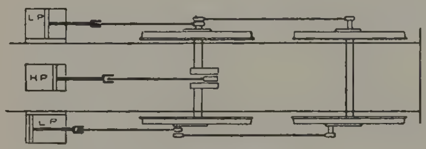
As the power of two-cylinder compounds was distinctly limited with the restriction imposed on the size of cylinders, they have practically become obsolete, although a few years back large numbers of such engines were built.

Having to add to the number of cylinders, with consequent increase in the weight of the mechanism of the engine, has introduced compromising factors into this problem of securing more powerful yet economical engines, and compounds are no longer favourites, although a perusal of the diagrams given (Fig. 10) will enable the reader to recognize the facility with which the articulated form of engine lends itself to compounding; hence, the retention of



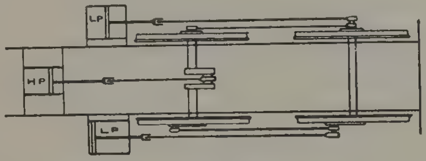
I. Two Cylinders

One about usual size (high pressure), one larger (low pressure).

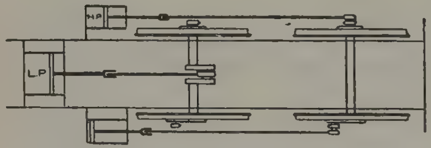


II. Three Cylinders

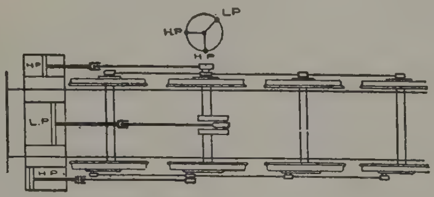
1. One high pressure, two low pressure.
(*a*) All cylinders drive one axle.



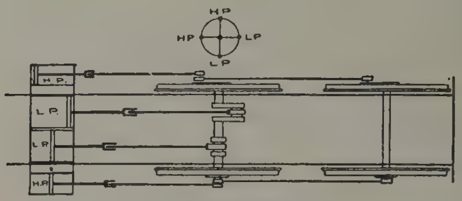
1. (*b*) Cylinders drive separate coupled axles.



2. Two high pressure, one low pressure.
(*a*) Cylinders drive separate uncoupled axles.

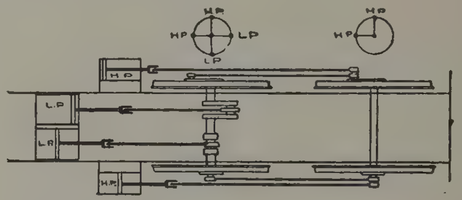


(*b*) All cylinders drive one axle.

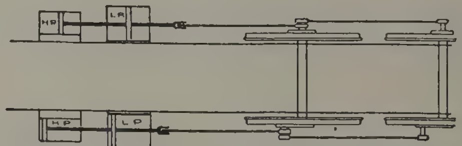


III. Four Cylinders

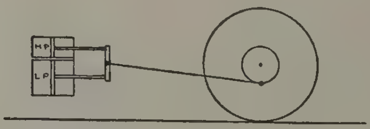
1. Balanced. All cylinders drive one axle. Also with low pressure cylinders outside.



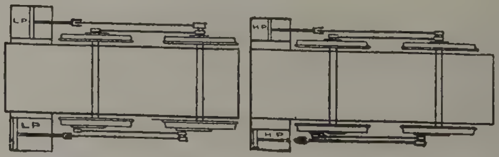
2. Divided and balanced. Separate coupled axles driven.



3. Tandem. Cylinders arranged in tandem. Two cranks only operated. In some cases, cylinders are connected and one slide valve employed for each pair of cylinders.



4. Superposed Cylinders. Two piston-rods connected to one crosshead and connecting rod, one slide valve for two cylinders. H.P. cylinder on top for passenger engines, L.P. above for goods engines.



5. Duplex or articulated. Two sets of coupled wheels (one or both in pivoted truck frames), one set driven by high pressure, one set by low pressure cylinders.

FIG. 10. CYLINDER ARRANGEMENTS FOR COMPOUND LOCOMOTIVES

the compound system in many of the very latest of these, notwithstanding the fact that they have also superheaters.

Recorded results of fuel consumption with compound locomotives performing similar duties to simple ones showed, when comparison was made, that with higher boiler pressures the compounds were the more economical, thus suggesting that the lower consumption of fuel might be more influenced by the increased pressure than by the fact that more extensive expansion was permissible.

An unique application of what may be termed ultra high pressure compounding, has recently been put into practice on the Continent. The engine has two boilers—in one steam is generated to a pressure of about 850 lb. per sq. in. This ultra high pressure steam is first used in a H.P. cylinder placed between the framing, from this it exhausts still at a comparatively high pressure, and is mixed with superheated steam received from the second boiler at a pressure of about 200 lb. per sq. in. This mixture then operates two outside cylinders, and finally escapes to the atmosphere, after losing a considerable portion of heat which is utilized for feed-water heating.

This combination is extremely ingenious, and theoretically there is much to be said for it ; it remains to be seen, however, if the returns in actual service will compensate for the extra cost in construction and maintenance.

PIONEERS OF ENGINEERING

By J. F. Corrigan, M.Sc., A.I.C.

LESSON XVIII

THE LAWS OF HEAT AND ENERGY

JAMES PRESCOTT JOULE (1818–1889)

Although not an engineer by profession, the name of James Prescott Joule must be included in our list of engineering pioneers. For Joule gave to the engineering and to the scientific world in general the outlines of two of the most comprehensive principles which science includes among its category of " laws."

The two famous principles with which the name of Joule is now inseparably connected are the Laws of the Conservation of Energy and the Mechanical Equivalent of Heat. The Law of the Conservation of Energy implies that energy, although it may be variously transformed, is entirely indestructible. Energy may be changed in form, but it can never be annihilated.

On the other hand, the principle of the Mechanical Equivalent of Heat holds that for a given amount of mechanical energy expended, an equivalent amount of heat is obtained. To quote Joule's own words : " The agents of Nature are, by the Creator's fiat, indestructible ; and whenever mechanical force is expended, an exact equivalent of heat is always obtained."

Joule worked out his doctrine of the Mechanical Equivalent of Heat by a method which will no doubt be very familiar to readers of this Educator. In principle, his apparatus was similar to the one illustrated on page 928. Two slowly descending weights were caused to rotate an arrangement of vanes contained in a copper vessel. The vessel was filled with water, and it was noted that the rise in temperature of the water caused by the friction of the rotating vanes was always exactly proportional to the amount of energy expended by the falling weights.

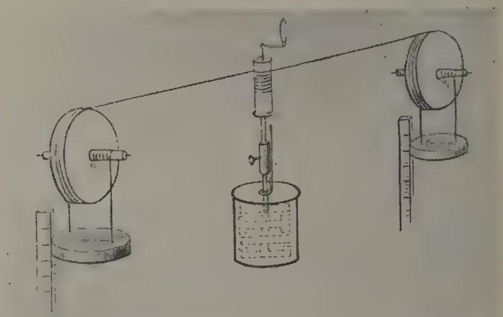
Joule also determined the amount of heat developed by the compression of air. He made a great number of other energy determinations, all of which were of the very greatest importance to the practical engineer and machine designer.

James Prescott Joule

He showed, in fact, that by carefully observing the various phenomena connected with the working of the steam and other motive engines, and by designing engines on principles dictated as a result of such observations, all heat engines could be greatly improved in actual working efficiency. Joule, therefore, although not a practical engineer himself, enunciated principles which were of the utmost value.

Now for a short notice of the man himself. Joule was a brewer by trade. Science was, for the most part, a hobby to him, and much of his scientific work was conducted in his spare time. Born at Salford, near Manchester in 1813, Joule received his early lessons in science from the famous John Dalton, of Manchester. After that period, he studied the advanced portions of physics and chemistry by himself. Every scientific investigator specializes in some particular type of scientific work, and Joule made the development of accurate methods of measurement, and the formulation of basic theoretical principles the aim of his scientific activities.

Without the doctrine of the Conservation of Energy, the Mechanical Equivalent of Heat, and the various other principles of energy transformation, modern science could hardly have reached its present position. Joule was not only a pioneer of engineering; he was a pioneer also in the domains of chemistry, electricity, physics, and mechanics.



Joule's Apparatus for the Determination of the Mechanical Equivalent of Heat

He died at Sale, near Manchester, in 1889, and the visitor to that city may see in the Town Hall a statue of him in an attitude of thought which is said to have been characteristic of the scientist. James Prescott Joule is a great name in the realms of theoretical science. It is one which will outlive that of many of the lesser celebrities.

MATHEMATICS FOR ENGINEERS

By W. G. Bickley, M.Sc.
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LESSON XIX

TRIANGLES—(*contd.*)

Cosine Rule (*contd.*). At the end of last lesson we had obtained two forms of the formula known as the cosine rule, a relation between the three sides of a triangle and one angle—

$$c^2 = a^2 + b^2 - 2ab\cos C \qquad . \qquad . \qquad (B)$$

and $$\cos C = \frac{a^2 + b^2 - c^2}{2ab} \qquad . \qquad . \qquad . \qquad (B')$$

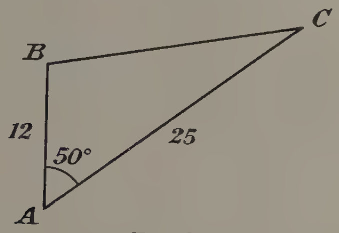


Fig. 84

We proceed to give some examples of the usefulness of these formulae.

Example 1. The vertical post of a crane is 12 ft. high, and the jib is 25 ft. long, inclined at 50° to the vertical. Find the length of the tie.

In Fig. 84, AB is the post, AC the jib, and BC the tie. We thus have a triangle in which $c = 12$ ft., $b = 25$ ft., and $A = 50°$. The formula for a is,

$$\begin{aligned} a^2 &= b^2 + c^2 - 2bc\cos A \\ &= 25^2 + 12^2 - (2 \times 25 \times 12)\cos 50° \\ &= 625 + 144 - 600 \times 0{\cdot}6428 \\ &= 383{\cdot}32. \end{aligned}$$

so $$a = \sqrt{383{\cdot}32} = \underline{19{\cdot}58 \text{ ft.}}$$

Example 2. Two forces of 5 lb. and 7 lb., act on a body in directions inclined at 65° to one another. Find the magnitude of their resultant, and the angle its line of action makes with the larger force.

Representing the forces by AB and AD (Fig. 85), the resultant is represented in magnitude and direction by the diagonal, AC, of the parallelogram $ABCD$. In the triangle ABD, we have $c = 7$, $a = 5$, $B = 180° - 65° = 115°$ so that

$$\begin{aligned} b^2 &= a^2 + c^2 - 2ac\cos B \\ &= 5^2 + 7^2 - 2 \times 5 \times 7\cos 115° \\ &= 25 + 49 + (70 \times 0{\cdot}4226) \end{aligned}$$

(remembering that $\cos 115° = -\cos(180° - 115°) = -\cos 65°$)

$$= 103{\cdot}58,$$

so $$b = \sqrt{103{\cdot}58} = 10{\cdot}178,$$

i.e. resultant is $\underline{10{\cdot}18 \text{ lb.}}$

To find the angle CAB (A in the triangle) we now use the sine rule—

$$\frac{\sin A}{a} = \frac{\sin B}{b}$$

giving $$\sin A = \frac{5\sin 65°}{10{\cdot}18}$$

whence, using logs, we find

$$\underline{A = 26°\,26'}$$

As a check upon the accuracy of the working, we may also find the angle C by the sine rule, when the three angles should add up to 180°.

This is an important example, for in mechanics we often need the resultant of two vectors, whether forces, velocities, accelerations, etc., and we always use the parallelogram rule. By reasoning similar to the above, we shall find that we have to deal with a triangle (half the parallelogram), two of whose sides represent the given forces (vectors) and the angle of which is the *supplement* of the angle between the

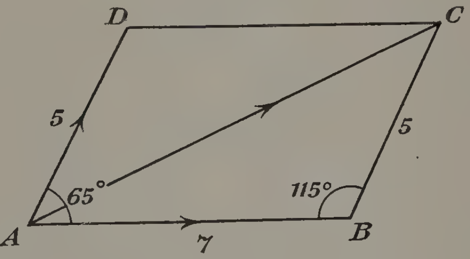


Fig. 85

forces (vectors). Thus, if P and Q are the forces (vectors), R their resultant, and θ the angle between them,

$$R^2 = P^2 + Q^2 - 2PQ \cos (180° - \theta),$$

or, using the rule for the cosines of supplementary angles, $\cos (180° - \theta) = - \cos \theta$,

$$R^2 = P^2 + Q^2 + 2PQ \cos \theta.$$

It will also be seen in this example, that by the combined use of the cosine and sine rules we are able to "solve" any triangle given the values of two sides and the angle between them. It was also pointed out at the end of the last lesson that the form (B') enables us to find the angles of a triangle if we know the sides. The following are instances of this—

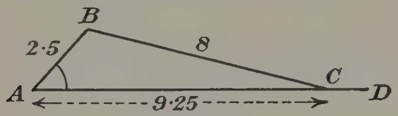


FIG. 86

EXAMPLE 3. Find the angles of a triangle with sides 4 in., 5 in., and 6 in.

If $a = 4$, $b = 5$, and $c = 6$,

$$\cos A = \frac{b^2 + c^2 - a^2}{2bc}$$

$$= \frac{25 + 36 - 16}{2 \times 5 \times 6}$$

$$= \frac{45}{60} = 0·75$$

so $\underline{A = 41° 25'}$

Similarly, $\cos B = \dfrac{36 + 16 - 25}{2 \times 4 \times 6} = 0·5625$

giving $\underline{B = 55° 46'}$

and $\cos C = \dfrac{16 + 25 - 36}{2 \times 4 \times 5} = 0·125$,

giving $\underline{C = 82° 49'}$.

(In such a case, it is possible to find the third angle by subtracting the sum of the first two from 180°, but it is always advisable to work out all three from the formula, when their adding up to 180° affords a check upon the accuracy of the working.)

EXAMPLE 4. The crank of an engine is 2·5 in. long, and the connecting rod is 8 in. long. Find the angle turned through by the crank when the piston travels through $\frac{1}{4}$ stroke from inner dead centre.

In Fig. 86, AB is the crank, AD the line of stroke, (D being the position of the crosshead at the end of the stroke, so that $AD = 8 + 2·5 = 10·5$ in.), and BC the connecting rod. $CD = \frac{1}{4}$ stroke $= 1·25$ in., so $AC = 9·25$ in. Consequently, in the triangle ABC,

$$\cos A = \frac{b^2 + c^2 - a^2}{2bc}$$

$$= \frac{9·25^2 + 2·5^2 - 8^2}{2 \times 9·25 \times 2·5}$$

$$= 0·6013$$

so $\underline{A = 53° 2'}$

We can also solve an example like (3) in Lesson XVIII by the use of the cosine rule. In that example we had to find the side c of a triangle in which $a = 3$, $b = 4$, and $A = 28°$. The cosine rule gives

$$a^2 = b^2 + c^2 - 2bc \cos A$$

$$9 = 16 + c^2 - 8c \cos 28°$$

or, using the value of $\cos 28°$, and transposing,

$$c^2 - 7·0632c + 7 = 0$$

The rule for quadratic equations gives

$$c = \frac{7·0632 \pm \sqrt{7·0632^2 - 4 \times 7}}{2}$$

$$= \underline{5·871} \text{ or } \underline{1·192 \text{ in.}}$$

as before.

We have now seen how the sine and cosine rules enable us to solve any triangles, and their use shortens the work. The sine rule is particularly convenient, since it only needs multiplication and division, and consequently logarithms can be used throughout. The cosine rule, however, contains additions and subtractions, so that it is not so convenient, since logarithms cannot be used throughout. If the numbers are small, so that most of the arithmetic can be done mentally, as in example 3 above, nothing better can be devised, but when the numbers are larger, the formula can be transformed into one that is better adapted to the use of logarithms. After obtaining some important relations between the sines, cosines, etc., of various angles in general we shall return to this point.

The student should now return to some of the early examples given in Lesson XVI, and see how the use of the sine and cosine rules lightens the work. In particular, Exercise 26, Nos. 6, 7, 8, 9, and 10. We also give further examples.

EXERCISE No. 30

1. Two sides of a triangle are 3 ft. and 4 ft., and the angle between them is 40°. Find the other side and angles of the triangle.

2. If the angle in (1) is 140°, solve the triangle.

3. Find the resultants of the following pairs of forces—
   (a) 6 tons and 2 tons, inclined at 60°.
   (b) 5 lb. and 3 lb., inclined at 135°.

4. Find the angles of the following triangles—
   (a) Sides 5 in., 6 in., and 8 in.
   (b) Sides 2 ft., 3 ft., and 4 ft.

5. The post, jib, and tie of a crane are 10 ft., 25 ft., and 18 ft. long, respectively, and the post is vertical. Find the inclinations of the tie and the jib to the horizontal.

6. The sides of an isosceles triangle are 5 in., 5 in., and 4 in. Find the vertical angle (*a*) by the cosine rule, and (*b*) by dividing the triangle into two equal right-angled triangles. Deduce from your method that $\cos A = 1 - 2\sin^2\frac{1}{2}A$.

7. Check the answers of Exercise 29, No. 4, by using the cosine rule.

Area of Triangle. We know that the area of a triangle is given by $\frac{1}{2}$ base × height. Thus, in Fig. 76 (Lesson XVIII)

$$\Delta = \tfrac{1}{2}bp.$$

But, as used in Lesson XVIII,

$$p = c\sin A,$$

so that

$$\Delta = \tfrac{1}{2}bc\sin A \qquad (C)$$

an important formula for the area of a triangle.

EXERCISE No. 30 (*contd.*)

8. Find the areas of the triangles in (1), (2), (4), and (6) above.

9. By dividing up an isosceles triangle into two equal right-angled triangles, obtain an expression for the area in terms of the sine and cosine of half the vertical angle, and the length of the equal sides. By comparing this with (*C*), deduce that $\sin A = 2\sin\frac{1}{2}A\cos\frac{1}{2}A$.

SUM AND DIFFERENCE FORMULAE

Two of the above examples, (6) and (9), show evidence that knowing the sine and cosine of half an angle, we can calculate the sine and cosine of the angle, or, putting it differently, that knowing the sine and cosine of an angle, we can calculate the sine and cosine of double the angle. These results are particular cases of general formulae connecting the ratios of the sum or difference of two angles with the ratios of the angles themselves. Without these formulae, and formulae derived from them, it is not possible to make much further progress in trigonometry, so we proceed to establish them. To do this we shall use a method depending upon the resolution of forces, but it *is* possible to read "projection of a line" for "component of a force" throughout the proof, and then get a purely geometrical proof.

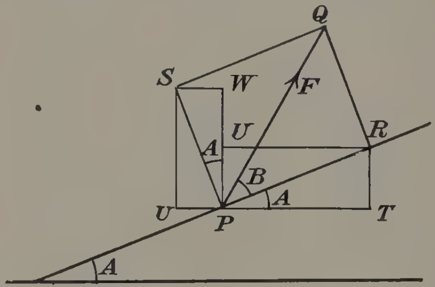


FIG. 87

Suppose (Fig. 87) a force F represented by PQ acts on a body at P on an inclined plane which makes an angle A with the horizontal, PQ making an angle B with the plane. Then the force makes an angle $(A + B)$ with the horizontal. Thus the vertical and horizontal components of F are $F\sin(A + B)$ and $F\cos(A + B)$ respectively. We may get the values of these components in another way, by first resolving the force into two components along and perpendicular to the plane, and then resolving these two components horizontally and vertically. The component along the plane, represented by PR, is $F\cos B$, and the component perpendicular to the plane, represented by PS, is $F\sin B$. The vertical component of PR, represented by TR, is $PR\sin A$, that is, $F\cos B\sin A$. The vertical component of PS is, since $\angle SPW = A$, $PS\cos A$, or $F\sin B\cos A$. Adding these two, we must get the vertical component of F, so that

$$F\sin(A + B) = F\cos B\sin A + F\sin B\cos A$$

Dividing by F and re-arranging, we have

$$\sin(A + B) = \sin A\cos B + \cos A\sin B \qquad (1)$$

Similarly, the horizontal component of PR is $PR\cos A$, that is, $F\cos B\cos A$, to the right; and the horizontal component of PS is $PS\sin A$, that is, $F\sin B\sin A$, to the *left*. Expressing the fact that these two give the horizontal component of F, and taking account of the directions, we have

$$F\cos(A + B) = F\cos B\cos A - F\sin B\sin A$$

or,

$$\cos(A + B) = \cos A\cos B - \sin B\sin A \qquad (2)$$

By an exactly similar process, taking the angle B *below* the plane, we can show that

$$\sin(\mathrm{A}-\mathrm{B}) = \sin\mathrm{A}\cos\mathrm{B} - \cos\mathrm{A}\sin\mathrm{B} \quad . \ (3)$$

and

$$\cos(\mathrm{A}-\mathrm{B}) = \cos\mathrm{A}\cos\mathrm{B} + \sin\mathrm{A}\sin\mathrm{B} \ . \ (4)$$

As numerical instances of these results, we will find the sine and cosine of 75°, and of 15°, by the use of these formulae, and the sines and cosines of 45° and 30°. From a right-angled isosceles triangle we can see that $\sin 45^\circ = \frac{1}{\sqrt{2}} = \cos 45^\circ$; and from half an equilateral triangle, we have $\sin 30^\circ = \frac{1}{2}$, and $\cos 30^\circ = \frac{\sqrt{3}}{2}$

By (1),

$$\begin{aligned}\sin 75^\circ &= \sin(45^\circ + 30^\circ)\\ &= \sin 45^\circ \cos 30^\circ + \cos 45^\circ \sin 30^\circ\\ &= \frac{1}{\sqrt{2}} \times \frac{\sqrt{3}}{2} + \frac{1}{\sqrt{2}} \times \frac{1}{2} = \frac{\sqrt{3}+1}{2\sqrt{2}}\\ &= 0{\cdot}9659\end{aligned}$$

Similarly, by (2),

$$\begin{aligned}\cos 75^\circ &= \cos(45^\circ + 30^\circ)\\ &= \cos 45^\circ \cos 30^\circ - \sin 45^\circ \sin 30^\circ\\ &= \frac{1}{\sqrt{2}} \times \frac{\sqrt{3}}{2} - \frac{1}{\sqrt{2}} \times \frac{1}{2} = \frac{\sqrt{3}-1}{2\sqrt{2}}\\ &= 0{\cdot}2588\end{aligned}$$

Using (3) and (4) in this way we find $\sin 15^\circ = 0{\cdot}2588$, $\cos 15^\circ = 0{\cdot}9659$. All these results agree with those found in the tables—in fact, it was by the use of these formulae that the tables were calculated in the first instance.

If in (1) and (2) we put $A = B$, so that $(A + B) = 2A$, we find

$$\sin 2A = \sin A \cos A + \cos A \sin A$$

i.e. $\quad \mathbf{\sin 2A = 2 \sin A \cos A}$

and $\quad \cos 2A = \cos A \cos A - \sin A \sin A$

or $\quad \mathbf{\cos 2A = \cos^2 A - \sin^2 A.}$

The latter can be put into two other forms by the use of the formula

$$\cos^2 A + \sin^2 A = 1.$$

From this, $\quad \sin^2 A = 1 - \cos^2 A$,

so that $\quad \cos 2A = \cos^2 A - (1 - \cos^2 A)$

or, $\quad \mathbf{\cos 2A = 2 \cos^2 A - 1.}$

Also, since $\quad \cos^2 A = 1 - \sin^2 A$

$$\cos 2A = (1 - \sin^2 A) - \sin^2 A$$

or $\quad \mathbf{\cos 2A = 1 - 2 \sin^2 A.}$

As a numerical instance of the use of these, if $\sin A = 0{\cdot}4$, $\cos A = \sqrt{1 - 0{\cdot}4^2} = \sqrt{0{\cdot}84} = 0{\cdot}9165$. Then,

$$\begin{aligned}\sin 2A &= 2 \sin A \cos A\\ &= 2 \times 0{\cdot}4 \times 0{\cdot}9165\\ &= 0{\cdot}7332.\end{aligned}$$

To find $\cos 2A$, the third of the above formulae is preferable, giving,

$$\begin{aligned}\cos 2A &= 1 - 2 \sin^2 A\\ &= 1 - 2 \times 0{\cdot}4^2\\ &= 0{\cdot}68.\end{aligned}$$

As a check, verify that these two values satisfy $\cos^2 2A + \sin^2 2A = 1.$

EXERCISE No. 31

1. If $\sin A = \frac{3}{5}$ and $\sin B = \frac{5}{13}$, find the values o $\cos A$, $\cos B$, $\sin(A + B)$, $\cos(A + B)$, $\sin(A - B)$, and $\cos(A - B)$.

2. If $\sin \theta = 0{\cdot}8$, find the values of $\sin 2\theta$ and $\cos 2\theta$, and then use these values to find $\sin 3\theta$ and $\cos 3\theta$.

3. Prove that $\sin 3\theta = 3 \sin \theta - 4 \sin^3 \theta$, and that $\cos 3\theta = 4 \cos^3 \theta - 3 \cos \theta$. Verify by putting $\theta = 30^\circ$ and $\theta = 45^\circ$.

4. If $\cos 2\phi = 0{\cdot}6$, find $\sin \phi$ and $\cos \phi$.

ANSWERS TO EXERCISE No. 29

1. 653·4 yd., 403·7 yd.
2. 1·57 and 2·16 in from end of stroke; 26·5 ft./sec.
3. 486·5 and 596·8 lb.
4. *Sin* $A = 1{\cdot}2856$, so A is impossible· 4·610 and 1·520 in.

MACHINE CONSTRUCTION AND DRAWING

By G. Wood, A.M.I.Mech.E.

LESSON XVIII

Comparative Strength. We have referred, in the last lesson, to the value of analysing the reference drawing. We may now proceed to consider the method of tabulating results in order to give them a comparative value.

Factor of Safety. The comparative strength of two machine details, subject to the same intensity of stress, depends upon the material of which the parts are made and the nature of the loading.

For purposes of reference, when comparing strengths of machine parts, we refer to the ratio the ultimate or breaking stress of the material bears to the working stress. This ratio is termed the *factor of safety*. The *factor of safety* embodies a margin for reliability, and also covers our ignorance of any indeterminate forces which may be induced under load.

In practice a common ratio of *factors of safety* is observed for certain conditions of loading. Further, consideration is given to the strength of a machine part which, in the event of a breakage, would endanger life, or where the breakage of a small part would in turn produce a general breakdown of the machine. In such cases the *factor of safety* has a very high value.

Practice. The student analysing the drawings of various machine details, with the object of obtaining practical values of factor of safety, will find that, while for a particular class of machine the factors of safety for the different kinds of loading will possess a common ratio, the values will as a whole be greater or smaller than the factors used in the design of other classes of machine. It is this relative value of the factors of safety used in the design of a class of machine which, together with the form of parts adopted, determines what in design is termed *practice*.

Analysis. We may now proceed, for an example in the method of obtaining a knowledge of practice, to analyse a few reference drawings.

Dead Loads. Fig. 1 shows the detail of the foot of a steel roof truss. The forces shown acting on this portion of the structure are a maximum and, apart from the variable load due to wind, the forces may be considered as dead loads.

Let f_t = tensile stress induced in the tie in pounds per square inch.

f = ultimate tensile strength of material in pounds per square inch.

A = minimum cross-sectional area of tie in square inches.

F = factor of safety.

$$f_t = \frac{T}{A} = \frac{8 \times 2240}{1{\cdot}08} = 16{,}600 \text{ lb.-sq. in.}$$

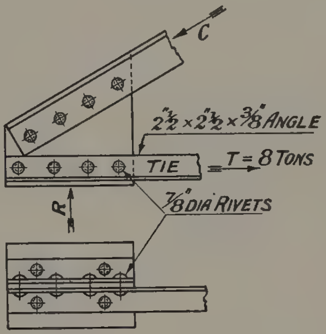


Fig. 1

The value of f for rolled steel sections is generally about 67,000 lb.-sq. in.

$$F = \frac{f}{f_t} = \frac{67{,}000}{16{,}600} = 4{\cdot}03.$$

The student is here advised to proceed to the analysis of further references of the above class of structure and compare the results obtained with the above factor.

Live and Intermittent Loads. Fig. 2 shows a piston-rod for a steam reciprocating engine. The rod is subject, at each stroke of the engine, to alternate tensile and compressive loads of T and C.

Let

P = maximum pressure on working side of piston in pounds per square inch.

p = minimum pressure on exhaust side of piston in pounds per square inch.

D = diameter of cylinder in inches.

d = diameter of piston-rod in inches.

d^1 = diameter of piston-rod at bottom of thread in inches.
l = length of rod between piston and cross-head in inches.
f_t = tensile stress induced in rod in pounds per square inch.
F_t = factor of safety in tension.
F_c = factor of safety in compression.

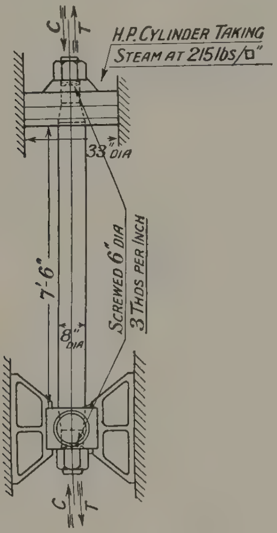


FIG. 2

Then, ignoring the inertia and weight of the moving parts,

$$T = \cdot 7854(D^2 - d^2)(P - p)$$

$$f_t = \frac{T}{\cdot 7854 d_1^{\,2}} = \frac{(D^2 - d^2)(P - p)}{d_1^{\,2}}$$

$$f_t = \frac{(33^2 - 8^2)(230 - 15)}{5\cdot 69^2} = 6806 \text{ lb. sq. in.}$$

$$F_t = \frac{63{,}000}{6806} = 9\cdot 25.$$

Proceeding to the case of the downward stroke

$$C = \cdot 7854 D^2 (P - p)$$
$$= \cdot 7854 \times 33^2 \times 215 = 184{,}000 \text{ lb.}$$

In finding a value for F_c it will be observed that the rod acts as a strut supporting the load C. The ratio of length to diameter of rod is sufficiently small to allow us to use the empirical Hodgkinson-Gordon formula for struts in finding a value for the factor of safety F_c.

The above formula gives the breaking load for mild steel circular sections as

$$W = \frac{36{,}000 \times \cdot 7854 d^2}{1 + \dfrac{l^2}{750 d^2}} \text{ lb.}$$

and substituting for d and l in the above example.

$$W = \frac{36{,}000 \times \cdot 7854 \times 8^2}{1 + \dfrac{90^2}{750 \times 8^2}} = \frac{36{,}000 \times 50\cdot 26}{1 + \cdot 169}$$

$$W = 1{,}550{,}000 \text{ lb.}$$

$$F_c = \frac{W}{C} = \frac{1{,}550{,}000}{184{,}000} = 8\cdot 4.$$

Shocks. Fig. 3 shows a propeller and shaft for a steam turbine driven ship. The twisting moment applied to the shaft is, for any given condition of running, of a constant value, but the resistance at the propeller is a variable quantity. When the ship is light or sailing in a heavy sea the propeller blades are not always immersed, with the result that the process of entering the water is in the nature of a shock.

In most vessels the distance between the propeller and the turbines is sufficiently large to prevent the variation in resistance to the twisting moment being transmitted to the turbines, so we may consider the following value of T to have a constant value, the maximum value at the propeller being covered by the large factor

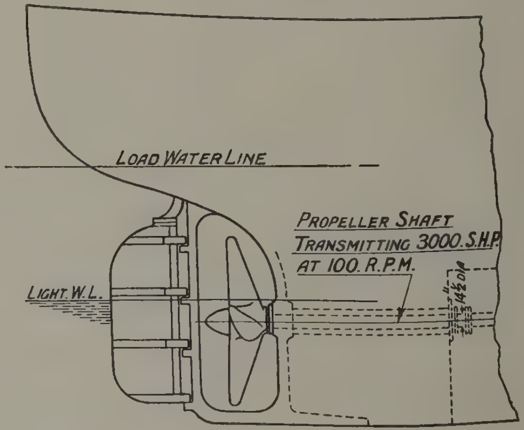


FIG. 3

of safety used in fixing the diameter of the shaft.

Let

H = horse-power transmitted through the shaft.

N = speed of propeller in revolutions per minute.

T = twisting moment applied to shaft at turbines in pounds-inches.

d = diameter of shaft in inches.

f_s = shear stress induced in shaft in pounds per square inch.

f = ultimate shear stress of material in pounds per square inch.

$$T = \frac{63{,}000\,H}{N} = \frac{63{,}000 \times 3000}{100} = 1{,}890{,}000$$

$$f_s = \frac{T}{\dfrac{\pi d^3}{16}} = \frac{1{,}890{,}000}{\dfrac{\pi \times 14{\cdot}5^3}{16}} = \frac{1{,}890{,}000}{598{\cdot}6} = 3160$$

$$F = \frac{f}{f_s} = \frac{50{,}000}{3160} = 15{\cdot}8.$$

Approximate Methods. We have already suggested that the factor of safety is in a large measure a factor of ignorance. There is yet another field of inaccuracy which it may usefully cover. It often happens that the precise stress induced in a loaded machine part can only be obtained with a somewhat lengthy and laborious calculation, when an approximate value of the stress may be found in a very simple calculation. Where the maximum ratio of the accurate to the approximate values is comparatively small and constant then the simple calculation may be used, the difference from the accurate being covered by a larger factor of safety.

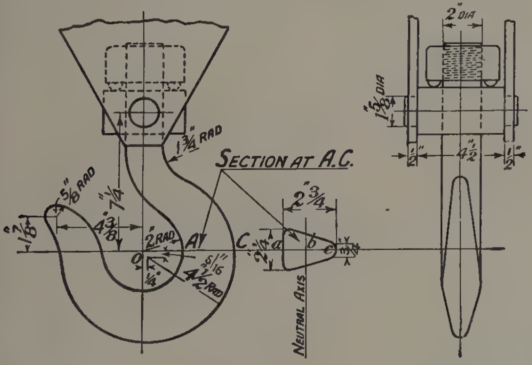


FIG. 4

The above method is suggested, not with a view to avoiding precision in estimating stresses, but rather to economize in time and so extend the field of analysis for useful data. We may add that the field of engineering is now so extensive that the designer is fully employed in applying the results of the research worker.

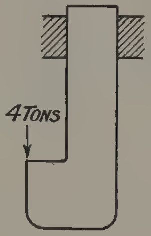


FIG. 5

Fig. 4, showing the hook and attachment of a 4-ton lifting block, provides an example of a bent beam. The calculation involved in finding the maximum stresses induced in the hook is not so simple as the case of the straight beam shown in Fig. 5. Further, it would be found that the stress induced in a straight beam, of the same proportion and loading, would be less than the stress in a curved beam. Also, the ratio of the two stresses so obtained would have an almost constant value for the proportions obtaining in practice.

Proceeding to the analysis of Fig. 4.

Let

W = maximum load on hook in tons.

I = moment of inertia of section of maximum B.M.

f_t = nominal tensile stress induced in tons per square inch.

f_c = nominal compressive stress induced in tons per square inch.

F = factor of safety.

Referring to Fig. 4 to find the position of the neutral axis of the section at A.C.

$$1{\cdot}5 \times 2{\cdot}75 \times ab = 2{\cdot}75 \times {\cdot}75 \times 1{\cdot}375 + {\cdot}75 \times 2{\cdot}75 \times \frac{2{\cdot}75}{3}$$

$$4{\cdot}125\,ab = 2{\cdot}84 + 1{\cdot}89 = 4{\cdot}73$$

$$ab = \frac{4{\cdot}73}{4{\cdot}125} = 1{\cdot}15.$$

$$\text{B.M.} = 4 \times 3{\cdot}15 = 12{\cdot}6 \text{ tons-in.}$$

$$I = \frac{\{2{\cdot}25^2 + (4 \times 2{\cdot}25 \times {\cdot}75) + {\cdot}75^2\} \times 2{\cdot}75^3}{36(2{\cdot}25 + {\cdot}75)}$$

$$= 2{\cdot}38 \text{ in.}^4$$

$$f = \frac{\text{B.M.} \times y}{I}$$

$$f_t = \frac{12{\cdot}6 \times 1{\cdot}15}{2{\cdot}38} = 6{\cdot}08 \text{ tons-sq. in. at } a.$$

Also the direct tensile stress

$$f = \frac{4}{4{\cdot}125} = {\cdot}96 \text{ tons-sq. in.}$$

Then the total nominal stress at *a*

$$f_t = 6{\cdot}08 + {\cdot}96 = 7{\cdot}04 \text{ tons-sq. in.}$$

$$\text{At } c \; f = \frac{12{\cdot}6 \times 1{\cdot}6}{2{\cdot}38} = 8{\cdot}5 \text{ tons-sq. in.}$$

and the total nominal stress at *c*

$$f_c = 8{\cdot}5 - {\cdot}96 = 7{\cdot}54 \text{ tons-sq. in.}$$

$$F = \frac{30}{7{\cdot}54} = 4.$$

Proceeding to the hook at the nut, we have the area at the bottom of the thread

$$= 2{\cdot}31 \text{ sq. in.}$$

$$\therefore f_t = \frac{4}{2{\cdot}31} = 1{\cdot}73 \text{ tons-sq. in.}$$

$$F = \frac{30}{1{\cdot}73} = 17{\cdot}3.$$

Each pin takes one-half the total load, ∴ the resistance to shear $= 2 \times 2{\cdot}07 = 4{\cdot}14$ sq. in., and the shear stress $f_s = \frac{4}{4{\cdot}14} = {\cdot}96$ tons-sq. in.

$$F = \frac{30}{{\cdot}96} = 31{\cdot}4.$$

The above value of f_s is absurdly low, and suggests a consideration other than strength for the large diameter of pin. As an alternative we may consider the bearing pressure on the pin.

Let p = bearing pressure on pin in pounds per square inch of projected area.

$$\text{Then } p = \frac{4 \times 2240}{2 \times 1{\cdot}625 \times {\cdot}5} = 5500 \text{ lb.-sq. in.}$$

The above value of p is sufficiently high to suggest that bearing pressure and not stress has been the limiting value in fixing the size of the pin.

EXERCISE

Make a drawing of a hook, similar in design to Fig. 4, to take a load of 6 tons. You may use the factors of safety obtained in the above example, and retain the same ratio of inner radius *OA* to outer radius *OC*.

ELECTRO-TECHNICS FOR MECHANICAL STUDENTS

By A. P. Young, O.B.E., M.I.E.E.

LESSON IX

DIRECT CURRENT GENERATOR OR DYNAMO

Principle of Operation. All modern generators of electricity depend for their operation on the principle of electro-magnetic induction. The first dynamo ever made was Faraday's disc dynamo, constructed in 1831, and shown diagrammatically in Fig. 48. Faraday provided two brushes, *A* and *B*, one bearing on the spindle and the other on the periphery of the disc, and in this manner utilized the voltage induced in the disc by its rotation in a magnetic field, to generate a current in an external circuit.

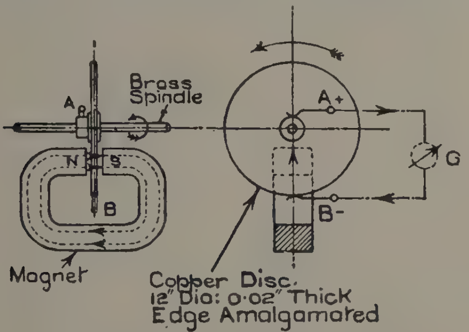


Fig. 48. Faraday's Disc Dynamo

The modern generator, a typical example of which is illustrated in Fig. 49, differs very considerably in construction from Faraday's disc dynamo, but nevertheless, the principle of operation is precisely the same. As now constructed, the generator comprises a rotary member, called the *armature*, and a fixed magnetic system known as the field magnets. The number of poles may be 2, 4, 6, 8, or an even higher multiple of two. Each pole projects radially from a cast steel yoke, to which it is securely fixed, and is surrounded by a magnetizing coil called the *field coil*. The poles are alternately N and S, the flux emanating from any N pole passing across the small air gap to the armature core and then dividing into two equal parts, each half of the flux flowing through the core and finally across the air gap to the adjacent S pole or poles.

The armature consists of a number of separate coils securely fitted into slots in the periphery of a laminated iron core. All the coils are connected in series, and the points of connection of adjacent coils are taken to segments of the commutator. Sets of brushes—usually equal in number to the poles—bearing on the commutator, collect the current from the rotating armature.

Voltage Induced in a Single Turn Coil. Let us now consider the simple case of a two-pole

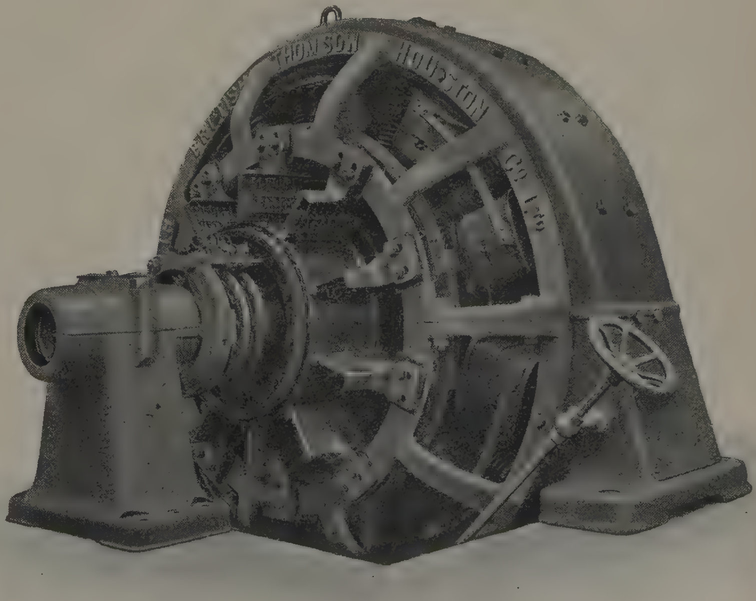


Fig. 49. Modern Twelve-pole Direct Current Generator

dynamo having a single turn coil on the armature, as illustrated in Fig. 50. The direction of rotation is assumed to be "clockwise" when viewed from the front end, and in the position

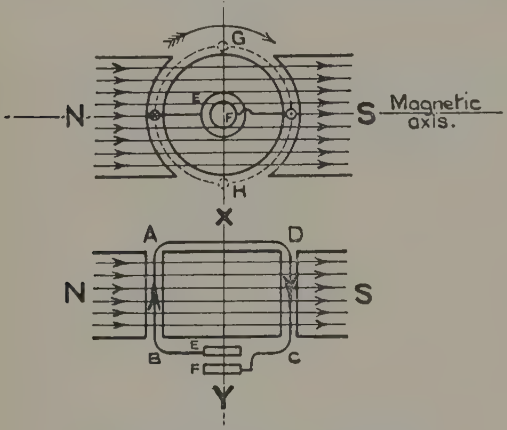


FIG. 50. SINGLE TURN COIL ROTATING IN TWO-POLE MAGNETIC FIELD

shown, the side *AB* will be *moving upwards* across the magnetic field, and the side *CD* *moving downwards* across another portion of the same magnetic field. Applying Fleming's right-hand rule, we find that the direction of the induced E.M.F. will be from *B* to *A* on the one side and from *D* to *C* on the other side. It should be particularly noted that the voltage induced in the complete coil is the sum of the induced voltages in the two portions *AB* and *CD*, thus making the end *E negative*, and the other end of the coil *F positive*. An E.M.F. is only induced in those portions of the coil—which are known as inductors—that lie parallel to the axis of rotation. The instantaneous voltage induced between the ends of the coil *EF*, will be twice the voltage induced in either inductor, due to its motion across the magnetic field.

A little consideration will show that the E.M.F.'s induced in the inductors are reversed twice during each revolution. When the inductors *AB* and *CD* have moved through 90°, they will then be in the positions shown by *G* and *H* respectively, and since they are then moving parallel to the magnetic lines, the induced E.M.F's. will be zero. Continuing the rotation, a further 90°, the inductors are again cutting lines of force at right angles, but in this case inductor *AB* is moving *downwards*, and inductor *CD upwards*, through the magnetic field. As the inductors are cutting the magnetic lines in a reverse direction to that shown, the E.M.F.'s induced therein will also be reversed. Thus, the induced E.M.F.'s will be *alternating* in character, and Fig. 51 shows graphically the induced voltage curve, assuming the speed of rotation

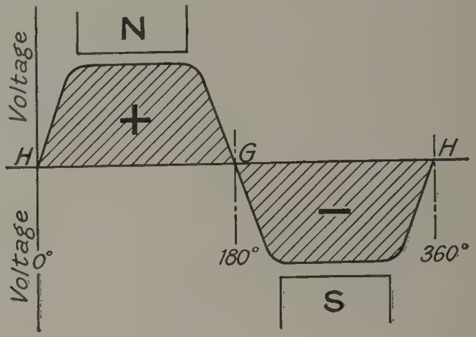


FIG. 51

to be constant. The zero points on this curve correspond to the positions *G* and *H* in Fig. 50.

The Function of the Commutator. It will be obvious that if an external circuit be connected to the single turn coil shown in Fig. 50, the current induced in that circuit will alternate twice during each revolution, in like manner to the induced E.M.F. In other words, an *alternating current* (A.C.) would be induced. If a unidirectional or *direct current* (D.C.) is required in the external circuit, some means must be provided for maintaining the terminals of the generator at the same polarity. This is done by means of a *commutator*, the function of which is to reverse the connections between the ends of the coil and the terminals of the generator, when the coil is in such a position that the E.M.F. would normally reverse its polarity. In the simple case under review, the commutator would consist of two segments of a ring, the ends of the coil being connected to these two

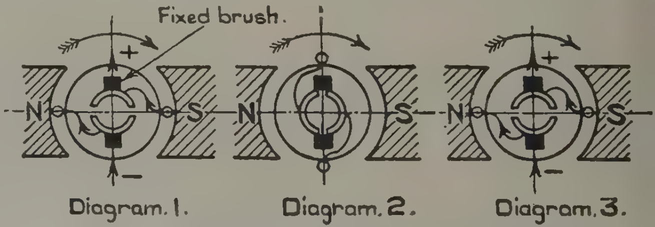


FIG. 52. DIAGRAMS ILLUSTRATING THE FUNCTION OF A COMMUTATOR

segments. In Fig. 52, three diagrams are shown illustrating the function of a commutator, from which it will be seen that the top brush is always positive and the bottom brush always negative. The induced E.M.F. at the brushes will, therefore, be uni-directional, although widely fluctuating in value from zero to maximum, as shown in Fig. 53, corresponding to the position of the inductors in relation to the magnetic field.

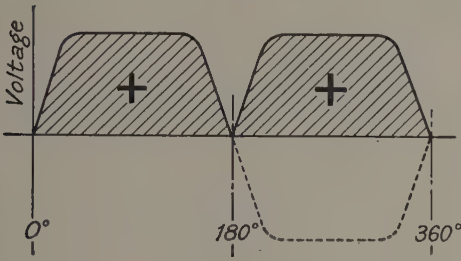


FIG. 53

The voltage fluctuations can be reduced by increasing the number of coils and, correspondingly, the number of segments. The coils are *connected in series*, and arranged around the periphery of the armature in such a manner that the voltage induced is *zero value in one coil at a time.* The brushes are also positioned so that any one coil undergoes commutation when the E.M.F. induced in it by rotation, is substantially zero (see diagram 2, Fig. 52). The instantaneous total E.M.F. induced in all the coils will be equal to the sum of the instantaneous E.M.F.'s induced in each coil, consequently in modern generators which have a large number of coils, the voltage fluctuations are, for all practical purposes, negligible, and so small that they cannot be detected with an ordinary voltmeter.

Armature Reaction in a Two-pole Generator. Armature reaction is the effect of the strong magnetic fields set up by the current flowing in the armature coils, and is a very important factor in the design of modern generators. Considering the case of a two-pole machine, as illustrated in Fig. 54, the armature conductors may be divided into two groups—

(*a*) Those lying between the vertical lines AB and CD. The current in these conductors will produce a magnetic field in the direction OE, in opposition to the main field, and in consequence will tend to reduce its value. This component (OE) is known as the *demagnetizing component.*

(*b*) Those lying outside the vertical line AB and CD. The current in these produces a magnetic field in the direction OF, which will have the effect of distorting the main field. This component (OF) is referred to as the *cross-magnetizing component.*

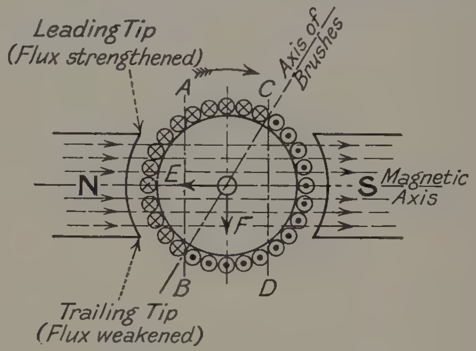


FIG. 54

The resultant effect of these two components is shown graphically in Fig. 55. The flux distribution at no-load—that is when no current is flowing in the armature—is indicated by the thick line. The magnetic field due to the demagnetizing and cross-magnetizing components are represented by OE and OF respectively, whilst the algebraical sum of these two curves and the no-load flux curve gives the resultant flux curve under load, indicated by the chain dotted line.

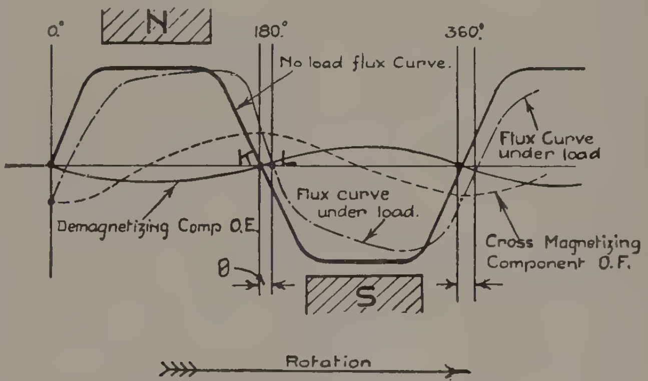


FIG. 55. EFFECT OF ARMATURE REACTION SHOWN GRAPHICALLY

The effect of armature reaction under load conditions is to distort the main field in the direction of rotation, thereby strengthening it under the leading pole tip and weakening it under the trailing pole tip. The zero point for the main flux is consequently displaced by the angle θ, i.e. from K to L. This angular displacement will increase with the load on the generator, and has an important bearing on commutation.

Commutation. In Fig. 56 is illustrated in a general way, an armature coil undergoing commutation. During the period the segments connected to the coil A are passing under the brush, the current in the coil has to die down to zero and build up again in a reverse direction. For sparkless commutation, the current in the short-circuited coil A should reverse its direction in accordance with the curve given in Fig. 57. Broadly speaking, this result is achieved if, during the period of commutation, the short-circuited coil is moving in a *reversing* field, which will be under the leading pole tip. The reversing field corresponds to some point on the flux curve in Fig. 55, *slightly beyond the zero point.*

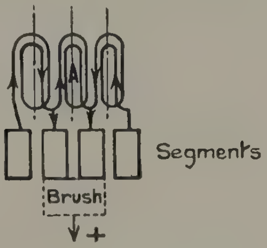


FIG. 56. COIL UNDERGOING COMMUTATION

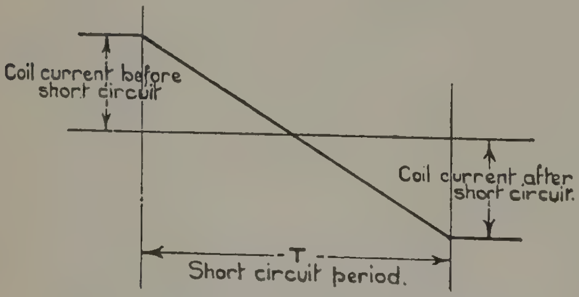


FIG. 57. CURRENT REVERSAL IN COIL WHEN COMMUTATION IS PERFECT

Commutating or Reversing Poles. It will follow from the foregoing that in order to obtain sparkless commutation, the brush positions must be adjusted to suit varying load conditions. This was certainly the case in the early dynamos, but, in modern machines, the necessity for this adjustment is obviated by the introduction of commutating or reversing poles. These are wound with magnetizing coils connected in series with the armature, and are, in general, placed midway between the main poles. In Fig. 58 is illustrated a two-pole generator with commutating poles. The magnetic field produced by the poles A and B is in the direction OF_1, and it is but a matter of design, to make this field equal in magnitude to the cross-magnetizing component OF of the armature coils. Since both fields are proportional to the armature current, the armature distorting effect is substantially neutralized under all load conditions.

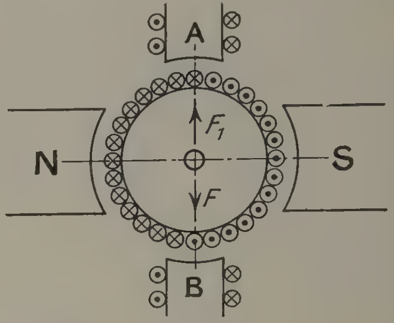


FIG. 58

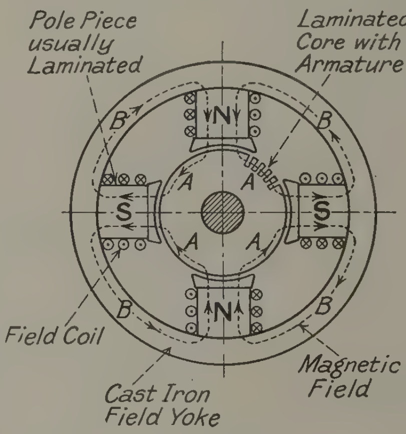

Pole Piece usually Laminated
Laminated Armature Core with Slots for Armature Coils
B
N
A
S
N
Field Coil
Cast Iron Field Yoke
Magnetic Field

FIG. 59

The Magnetic System. The main essentials of the magnetic system, together with the materials commonly employed, are indicated in Fig. 59. The armature core is built up of thin iron laminations of good permeability, which are insulated from one another by a thin coating of insulating varnish. This is done to reduce to a minimum the deleterious effects of eddy currents in the rotating iron mass. On large generators it is now general practice to build up the pole pieces of laminations somewhat thicker than those used for the armature.

The distribution of the flux is indicated by the chain lines, and the flux passing through the section A of the armature core is substantially half the total flux flowing through any one-pole piece. This is equally true of the flux in section B of the field yoke.

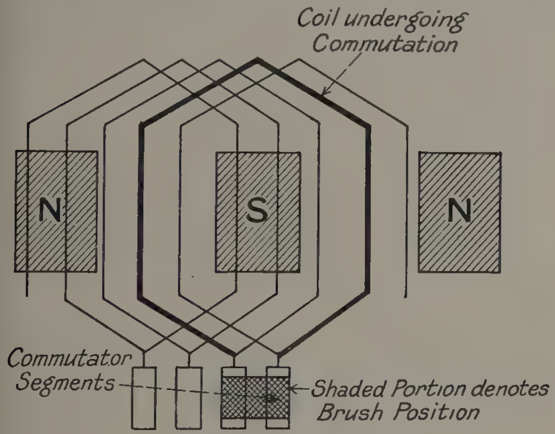

Coil undergoing Commutation
N
S
N
Commutator Segments
Shaded Portion denotes Brush Position

FIG. 60. LAP WINDING

Armature Windings. There are two types of winding employed in modern generators, viz.: *Lap Winding* and *Wave Winding*.

(*a*) *Lap Winding*. Lap winding, as illustrated in Fig. 60, is generally employed on machines of large output, and has as many circuits from positive to negative brushes as there are poles. Each circuit is in parallel, and the total output of the machine is equally divided in the various circuits. Thus, in a four-pole generator giving 800 amp., each circuit would be delivering 200 amp. The generator voltage can be calculated from the formula—

$$E = \frac{\Phi S n}{10^8} \quad . \quad . \quad . \quad . \quad . \quad (23)$$

Where Φ = flux per pole.
S = total number of armature inductors.
and, n = revolutions per second.

(*b*) *Wave Winding*. A wave winding (see Fig. 61) has only two circuits through the armature, irrespective of the number of poles. It is generally adopted for machines of low output and high voltage. The generator voltage in this case is calculated from the formula—

$$E = \frac{x}{2} \frac{\Phi S n}{10^8} \quad . \quad . \quad . \quad (24)$$

where, x = number of poles.

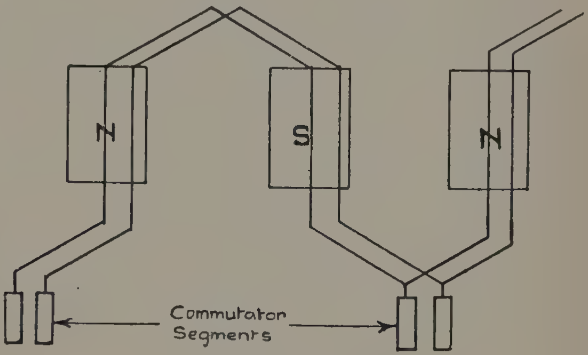


FIG. 61. WAVE WINDING

Commutators and Brushes. Commutators are built up of hard copper segments substantially of the section shown in Fig. 62, and are insulated from one another by mica strips. These segments are usually held together by steel clamping rings insulated with mica, which fit into the grooves A and B. On small generators using commutators up to about 6 in. diameter, it is American practice to use a moulded commutator in which the insulation between the segments and the supporting metal sleeve is a synthetic moulded composition.

The brushes which bear on the commutator are made of carbon, moulded to shape under great pressure. Copper is sometimes added to increase the conductivity of the brush. Carbon brushes, owing to their high resistance, materially assist commutation. During the first period of commutation, when the current in the short-circuited coil is falling from a maximum to zero, the high resistance of the carbon helps the current to die away.

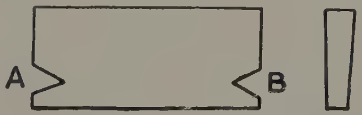


FIG. 62. COMMUTATOR SEGMENT

JIGS AND TOOLS

By Leo Kelsey, A.M.I.P.E.

LESSON VIII

GRINDING FIXTURES

We do not propose to enter here into that class of grinding which is concerned with imparting a rough or smooth finish to castings, stampings, forgings, etc., which is generally performed on large diameter grinding wheels mounted horizontally, with a table on which to rest the work, which is manipulated by the operator.

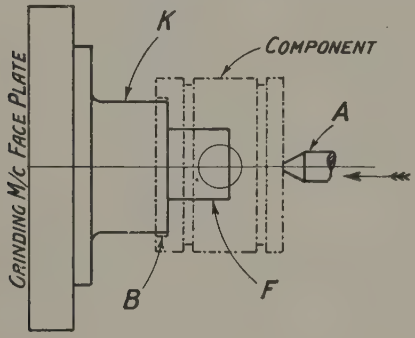


Fig. 70

Because grinding is, as a rule, the last process which is performed on a component, the need for locating and securing devices which will not distort the latter becomes acute and must, therefore, receive the most careful consideration. Our previous statements regarding ease of loading and removing work, cleaning locating surfaces, etc., remain true respecting this class of fixture.

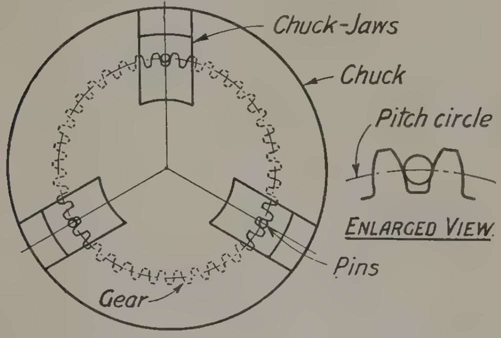
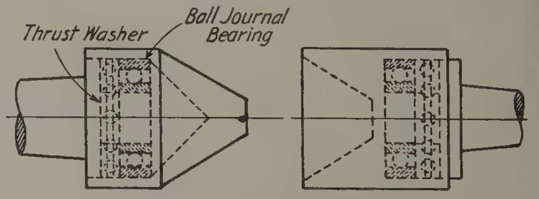


Fig. 71

Automobile pistons are ground by supporting them, as shown in Fig. 70. The centre *A* is spring-loaded, and the recess which has been turned in the piston at *B* is a push-fit on the spigot *K*, which drives the work by means of a projection at *F*, which loosely engages the machined surface between the gudgeon pin bosses. Expanding chucks should not be used here, because they enlarge and distort the component. When gear wheels have been hardened, but the teeth are not to be ground, it is necessary, for silent running, to grind the bore from the pitch circle of the teeth. This is done by inserting pins in the three jaws of a self-centring chuck, as in Fig. 71. These pins are of such a diameter that when they close on the gear, they engage the latter on the pitch circle. This is shown in the enlarged view in the same illustration. Should the total number of teeth in the gear to be ground, not be a multiple of three, the nearest tooth space should be used in order that the gear is clamped at three points, which are as equidistant as is possible.

In grinding tubes, which may be soft, a floating centre is used. The principle of this centre is, that the part of the centre which is in contact with the component revolves with the latter; the rotary load being taken by a suitable ball-bearing, while the thrust is taken by a ball-thrust washer. These centres are always used where the diameter is large, and the friction likely to be considerable. An illustration of a male and female revolving or "dead" centre is shown in Fig. 72.



Male Centre Female Centre

Fig. 72

Cam Grinding. The production method of grinding automobile camshafts, is to mount the work carrying centres upon trunnions in such a manner that the work is made to follow the same contour as a master cam mounted on the machine, and acting directly on to the work carrying device. The master cam is always made as large as possible, as in this manner the unavoidable errors which have occurred during its manufacture will be minimized by means of the inverse leverage thus produced. The method of achieving this is shown diagrammatically in Fig. 73. Constant rise cams can be generated by means of special attachments on the universal grinding machine. In the grinding of special cams in the tool-room, it is safer to make a large master cam in mild steel and rig this on to the universal machine, using a follower to impart its contour to the smaller cam blank which has to be cut.

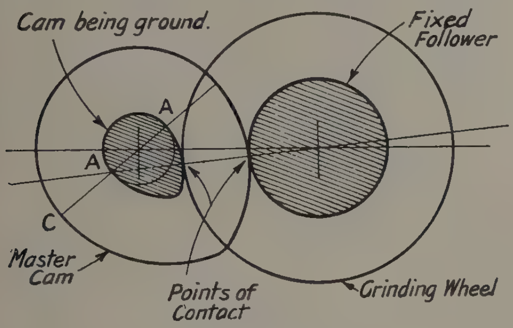


FIG. 73

Thread Grinding. We do not propose to enter into a detailed discussion of thread grinding, as this branch is exceedingly specialized, and should detailed information be desired, it can be obtained from the catalogues of the manufacturers of the special machine required. However, it is necessary that the general principle of thread grinding be understood, in the event of it being required in the tool-room.

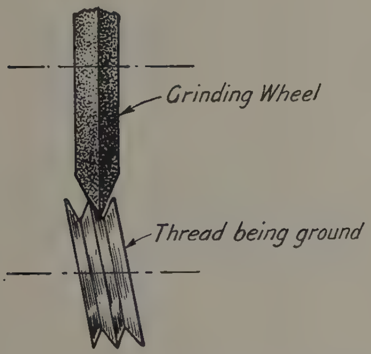


FIG. 74

A diagrammatic view of the arrangement is shown in Fig. 74. The grinding wheel is sharpened to the angle of the thread to be ground in the case of a vee thread, and frequent retrimming is necessary to ensure accuracy. A saucer wheel is generally used for the grinding of square threads.

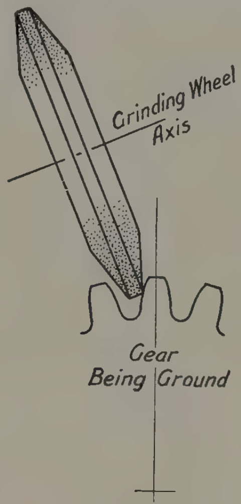


FIG. 75

Gear Grinding. We must apply much the same remarks to gear grinding as to thread grinding. Special machines are necessary to carry this out as on a production basis, and should it be required to grind a set of gears in the tool-room, it will be found very troublesome, as to do the job satisfactorily special machines are, for all practical purposes, indispensable. However, a diagrammatic view of the way in which the wheel is applied to the gear is shown in Fig. 75. A type of saucer wheel is sometimes

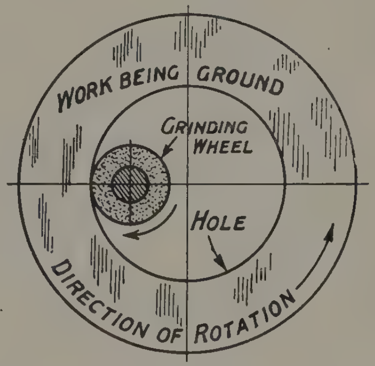


FIG. 76

used for this class of work. The grinding wheel has the same section as an involute rack tooth of the same pitch and pressure angle as the gear being ground. The rapidly rotating grinding wheel is moved in the direction of its axis, whilst the gear is rotated at such a rate that it meshes with the gear wheel as if the latter were a rack.

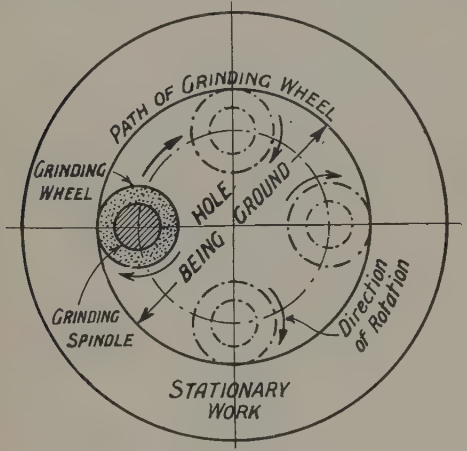


Fig. 77

Hole Grinding. There are two distinct methods of producing ground holes. In the first (Fig. 76), the grinding wheel rotates at a very high speed, and the work is revolved at a much lesser speed in the opposite direction. This method of grinding holes is quite satisfactory if the diameter is not larger than 1 to 2 in. However, with large diameters, the method shown in Fig. 77 is resorted to. This is known as cylinder grinding.

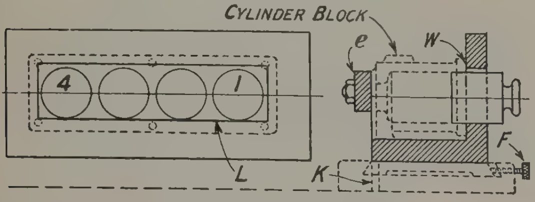


Fig. 78

The grinding wheel, in this case, revolves at a high speed, but also in a planetary motion about the centre of the hole to be ground. The result is that the grinding wheel spindle travels round the work about the latter's centre, the work, of course, remaining stationary. The feed for increasing the bore of the hole is obtained by increasing the throw of the eccentric to which the grinding wheel spindle is attached. Machines made on this principle can produce holes which are within very close limits, both as to circularity and diameter. Internal combustion engine cylinders are ground on machines of this type in the fixture illustrated in Fig. 78. The method adopted in lining up the bores is to put two plugs in the two end cylinders, Nos. 1 and 4, and rest these plugs on the straight edge *L*, as shown in the fixture. The clamp *C* presses the cylinder block against the face *W*, thus holding the block in position. Before grinding, the plugs are removed. To obtain correct centre distance between the cylinders, the fixture is mounted on slides and located by means of the plunger *K*, and clamped by means of the screw *F*. If only a small quantity is required, however, the table of the machine can be traversed and located by means of adjustable stops.

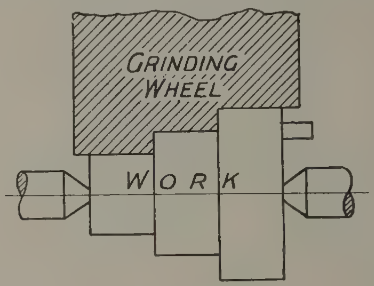


Fig. 79

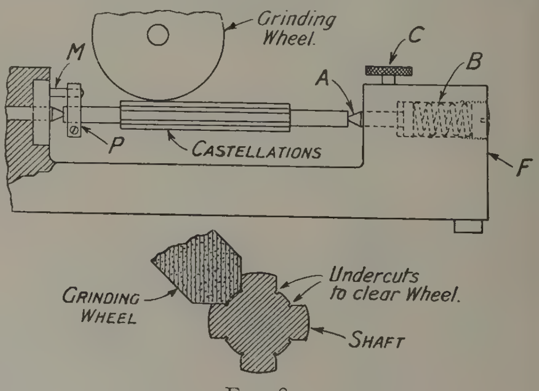


Fig. 80

Form Grinding. This method is the grinding analogy of form milling. The principle is that a broad grinding wheel is shaped to the contour of the work to be ground and fed directly on to the latter, no traverse in the case of cylindrical work being used. It might be thought, at first glancing at Fig. 79 (which shows a grinding wheel grinding a shaft of three diameters), that frequent trimming of the wheel would be necessary to ensure accuracy. This is not the case, however, as if the right grade wheel has been chosen, and the work and the wheel are run at the correct speed, very little trimming is necessary. The limit on this class of work is only reached by the width of the wheels, as these become very difficult to manufacture when about 10 to 12 in. wide. The depth of the form also is a limitation, due to the circumferential velocities which are met with when contours exceeding 1 in. in maximum depth have to be ground. Instead of using very broad wheels, it is possible to group two or more different diameter grinding wheels on the same spindle. However, this is not always so economical as using a broad wheel.

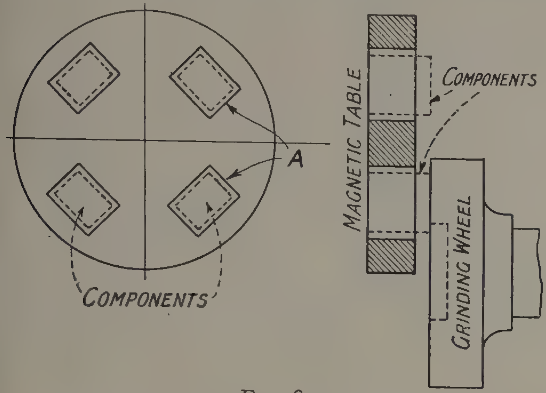


FIG. 81

A good example of form wheel grinding is shown in Fig. 80, where a castellated gear-box shaft is being ground in a horizontal surface grinding machine. The sides of the castellations, as well as the surface, are ground at one setting, a simple indexing fixture, supporting the work upon centres, being used. In this fixture the moving centre A is pressed against the component by means of the spring B, clamping being effected by means of the split housing F, and the clamping handle G. The work is gripped by the carrier at P, which engages with the dog M for indexing purposes.

Rotary Surface Grinding Machines. An example of a fixture for use on this type of machine is shown in Fig. 81. It consists of a plain ring with apertures cut in at A, to loosely fit the components, which in this case are bronze distance bosses. The ring is secured by the magnetic table of the machine. As the bronze is non-magnetic, the fixture is necessary to secure it. However, should it be a steel boss, it would still be economical to use such a fixture, as the rapidity with which it can be located on the table, in the correct position to come within the path of the grinding wheel, is ample justification for its use.

Another type of fixture is shown in Fig. 82, which gives the fixture for grinding the faces of gear wheels. These are held by means of the split spigots W, which are expanded in the bore of the component by means of the taper screw H. A plain screwdriver is used to operate this fixture.

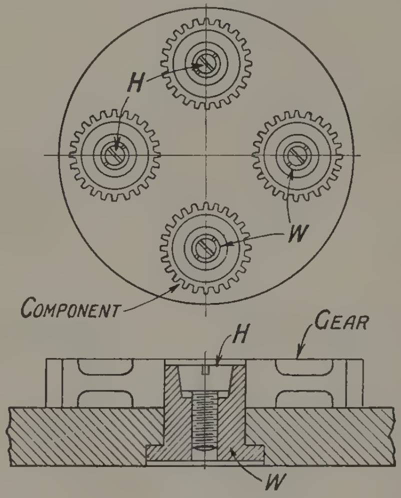


FIG. 82

POWER TRANSMISSION BY BELTING

By W. G. Dunkley, B.Sc.

LESSON VIII

WORKED EXAMPLES

We shall now continue to present a number of typical examples for your study and reference when you have a similar case to deal with. In the previous examples we took no account of centrifugal force.

Allowance for Centrifugal Force. We must remember that we have a stress in the belt, due to centrifugal force, and this should be allowed for. The simplest way to deal with this is to determine what stress per square inch is produced by the centrifugal force, and then deduct this amount from the allowable stress per square inch.

Example 12. Determine the deduction from the allowable stress per square inch for a belt running on a pulley 24 in. in diameter, which makes 600 r.p.m. to allow for the stress due to centrifugal force.

From Lesson VI we have

$$S = \frac{WV^2}{3600 \times g \times t} \text{ lb./sq. in.}$$

We can now put this into a more simple form—

W = weight of 1 ft. of belt 1 in. wide.
Let W_1 = weight of 1 ft. of belt 1 sq. in. area.

$$\text{Then } W = W_1 \times \text{thickness}$$
$$= W_1 \times t \times 1.$$

$$\text{Then } S = \frac{W_1 \times t \times V^2}{3600 \times g \times t}$$
$$= \frac{W_1 \times V^2}{3600 \times g}$$

Now we shall take W_1 to have a constant value of 0·5 lb.

$$\text{Then } S = \frac{0{\cdot}5 \times V^2}{3600 \times 32{\cdot}2} = \frac{V^2}{230{,}000}$$

In our example the value of V is given by

$$V = \frac{24}{12} \times \pi \times 600 = 3760 \text{ ft. per min.}$$

$$S = \frac{3760 \times 3760}{230{,}000}$$
$$= 62 \text{ lb./sq. in.}$$

Thus, instead of using an allowable stress of 320 lb./sq. in. for a laced single belt, we should have to allow only 320 – 62 = 258 lb./sq. in. You will notice that the same deduction is made irrespective of whether the belt is single, double, etc. Thus, for a laced double belt we would allow 240 – 62 = 178.

Example 13. In example 11 we neglected the allowance for centrifugal force. What should we have allowed?

$$\text{In that case } V = \frac{12}{12} \times \pi \times 600 = 1880.$$

$$\text{Then } S = \frac{1880 \times 1880}{230{,}000} = 15{\cdot}4.$$

This may be considered negligible. Up to about values of V = 3000 ft. per minute, the stress due to centrifugal force may be neglected, but for higher values it should be considered.

Belt Size Examples

Example 14. A 20 h.p. motor runs at 800 r.p.m. Assuming a pulley 12 in. diameter, determine a suitable belt.

Take angle of lap 170°. $\mu = 0{\cdot}4$.

This is generally how the proposition arises in practice.

We will, however, first need to determine $\frac{T}{t}$.

$$\text{Log. } \frac{T}{t} = 0{\cdot}007578 \times {\cdot}04 \times 170$$

$$\text{Then } \frac{T}{t} = 3{\cdot}28.$$

Now we can determine P from expression (4)

$$\text{h.p.} = \frac{P \times D \times N}{126{,}000}$$

$$\therefore \quad P = \frac{H \times 126{,}000}{D \times N}, \text{ where } H = \text{horse-power}$$
$$= \frac{20 \times 126{,}000}{12 \times 800} = 263 \text{ lb.}$$

$$\text{Now } P = \frac{\left(\frac{T}{t} - 1\right)}{\frac{T}{t}} \times T \text{ from (5)}$$

$$= \frac{(3{\cdot}28 - 1)}{3{\cdot}28} \times T$$

$$= \frac{2{\cdot}28}{3{\cdot}28} \times T$$

$$\therefore\ T = \frac{3{\cdot}28}{2{\cdot}28} \times P$$

Substituting our value of P, just determined,

$$T = \frac{3{\cdot}28}{2{\cdot}28} \times 263 = 380 \text{ lb.}$$

We have now to determine what section of belt we require to carry the above tension. Let us suppose a laced joint. Let x = width of belt required. For a single belt we can allow 320 lb./sq. in.

Then $x \times t \times 320$ = tension allowable = 380

$$x = \frac{380}{320 \times \frac{3}{16}}, \text{ (putting } t = \tfrac{3}{16} \text{ in.)}$$

$$= 6{\cdot}35 \text{ in.}$$

We should then use, say, a $6\frac{1}{2}$ in. belt.

For a laced double belt we allow 240 lb./sq. in.

$$x \times t \times 240 = 380$$

$$x = \frac{380}{240 \times \frac{3}{8}}, \text{ (putting } t = \tfrac{3}{8}.)$$

$$= 4\tfrac{1}{4} \text{ in. approx.}$$

Order of Procedure. For these cases, where we require to determine the belt size, the procedure is—

1. Find $\dfrac{T}{t}$

2. Find the value of P from the expression

$$P = \frac{H \times 126{,}000}{D \times N}$$

3. Determine T from $T = \dfrac{\dfrac{T}{t}}{\left(\dfrac{T}{t} - 1\right)} \times P.$

4. Decide on an allowable stress per square inch for the belt, allowing for the necessary reduction for the stress due to centrifugal force where necessary.

5. Determine size of belt.

EXAMPLE 15. We desire to transmit 50 h.p. from a pulley 18 in. diameter running at 900 r.p.m. Angle of lap 175°. $\mu = 0{\cdot}4$. Determine a suitable belt.

1. Log. $\dfrac{T}{t} = 0{\cdot}007578 \times 0{\cdot}4 \times 175$

Then $\dfrac{T}{t} = 3{\cdot}4$

2. $P = \dfrac{H \times 126{,}000}{D \times N} = \dfrac{50 \times 126{,}000}{18 \times 900}$

$$= 390 \text{ lb.}$$

3. $T = \dfrac{\dfrac{T}{t} \times P}{\left(\dfrac{T}{t} - 1\right)} = \dfrac{3{\cdot}4}{2{\cdot}4} \times P$

$$= \frac{3{\cdot}4}{2{\cdot}4} \times 390$$

$$= 550 \text{ lb. max. belt tension.}$$

4. We now require to find the stress S due to centrifugal force.

Belt velocity $V = \dfrac{18}{12} \times \pi \times 900 = 4250$ ft. per minute.

Stress $S = \dfrac{V^2}{230{,}000} = \dfrac{4250 \times 4250}{230{,}000}$

$$= 80 \text{ lb./sq. in. approx.}$$

5. (*a*) *Single cemented belt:* Stress per square inch allowable 400 lb. Allowing for centrifugal force, the stress must be reduced to 400 – 80 = 320 lb./sq. in.

Then $x \times \frac{3}{16} \times 320 = T = 550$

$$x = \frac{550 \times 16}{320 \times 3} = 9{\cdot}2 \text{ in.}$$

(*b*) *Single laced belt.* Allowable stress 320. We use, therefore, 320 – 80 = 240 lb./sq. in.

$$x \times \tfrac{3}{16} \times 240 = T = 550$$

$$x = \frac{550 \times 16}{240 \times 3} = 12{\cdot}2 \text{ in.}$$

(*c*) *Double cemented belt.* Stress allowable 300. We use, therefore, 300 – 80 = 220.

$$x \times \tfrac{3}{8} \times 220 = 550$$

$$x = \frac{550 \times 8}{220 \times 3} = 6{\cdot}7 \text{ in.}$$

(*d*) *Double laced belt.* Stress allowable 240. We use, therefore, 240 – 80 = 160.

$$x = \frac{550 \times 8}{160 \times 3} = 9{\cdot}1 \text{ in.}$$

Now decide which to adopt.

STUDY. Determine the belt to transmit 30 h.p. Pulley 15 in. diameter, r.p.m. 920, lap 180°. $\mu = 0{\cdot}4$. Allow for centrifugal force.

Ans. (*a*) 6 in., (*b*) 7·8 in., (*c*) 4·2 in., (*d*) 5·6 in.

GAUGING AND INSPECTION

By Joseph G. Horner, A.M.I.Mech.E.

LESSON III

GAUGES

Combinations of Precision Blocks. This method differs from that in which fixed cylindrical and snap gauges are employed. It is the making up of a dimension, or its checking by the combination of several blocks, each of which is of a definite thickness, and within limits of at least $\frac{1}{10000}$ in. They eliminate the errors of the human element, in consequence of which different men will obtain various results from calipers and rule, and they do not require frequent inspection and correction as even the best micrometer tools do, consequent on wear and rough usage. They are a unique product of the Swedish C. E. Johansson Co., Ltd., Coventry. They are not of the same class as the standard reference blocks previously described, in which a single block has one dimension, or two, and is not related to any other. The Johansson blocks give sizes advancing by very minute increments, expressed in decimals, by the selection and addition of which many thousands of combined dimensions can be obtained. It is a remarkable fact that cumulative errors are unknown, for however numerous the blocks selected are, they give the total dimension correctly if tested with a gauge. The blocks are therefore self-checking. Their utilities include checking the accuracy of snap gauges, and internal limit gauges, in resetting adjustable gauges to new limits, and to correct them for wear, to set heights for sine bars, to measure dies and jigs, and for general inspection. The blocks are made in sets of standard measures, and for limits, and in English and metric dimensions. They are made in four qualities or degrees of accuracy, two for standard use or reference, and two for shop services. The highest tolerance in the first is 0·000002 in. per inch, that in the lowest is 0·000012 in. per inch. In metric measures, the highest is 0·00005 mm. per 25 mm., that in the lowest is 0·0003 mm. per 25 mm.

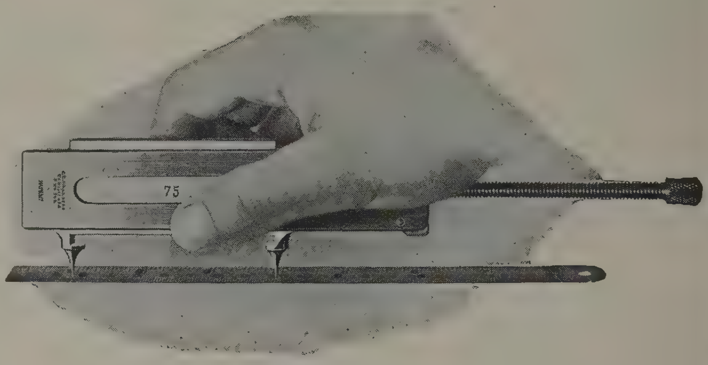


Fig. 16. Graduations Checked with Centre Points in Holder

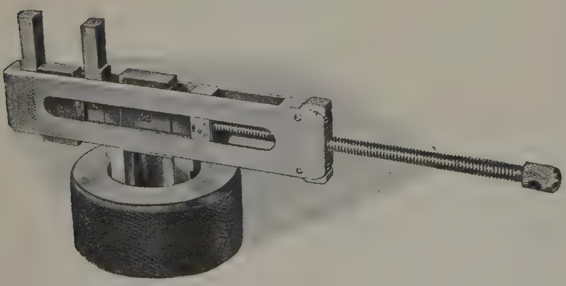


Fig. 15. A Holder, with Blocks and Jaws Gauging a Bore

Building up Dimensions. Johansson blocks for general or standard use are made in two sets in English sizes, and in five sets in metric sizes. With these sets, dimensions can be built up from 0·1 in. or from 1 mm. to the combined length of all the blocks. The difference between the sets consists in the number of blocks. In the larger sets it is possible to build up the desired dimensions with a minimum number of blocks. From the smaller sets, a larger number of blocks must be used to build up the same dimension. The largest standard set contains 81 blocks in four series. Nine in the first series range from 0·1001 to 0·1009 in., forty-nine in the second series range from 0·101 to 0·149 in., nineteen in the third series from 0·050 to 0·950 in., and four in the fourth series from 1 to 4 in. And these are supplied in three degrees of accuracy. The blocks in the first series will divide up the spaces between those of the second series, and those of the third and fourth series can be divided up by the first and second series. Over 80,000 different sizes can be obtained, accurate at 62° F.

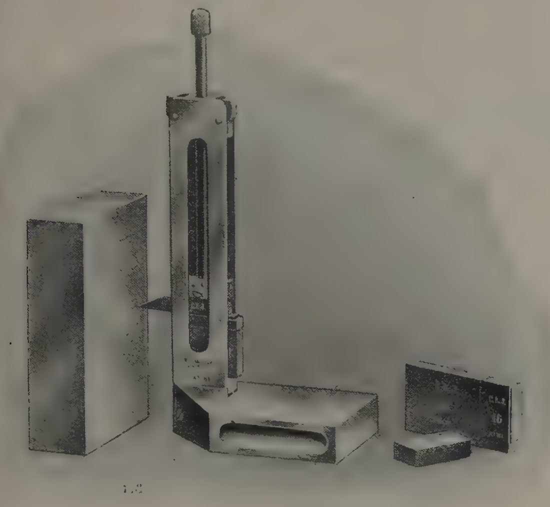


FIG. 17. HOLDER WITH SCRIBER POINT SET FROM A BASE BLOCK

The dimension required is most readily arrived at by eliminating the smallest decimal, working from right to left, and finishing with the largest block.

Thus, required—

1·248 in.	2·6483 in.	0·8391 in.
·148 ,,	·1003 ,,	·1001 ,,
·100 ,,	·1480 ,,	·1090 ,,
1·000 ,,	·4000 ,,	·1300 ,,
	2·0000 ,,	·5000 ,,
1·248 ,,	2·6483 ,,	·8391 ,,

The limit gauge blocks, made also in English and metric measures, contain eighteen blocks, in two series, nine having minus or negative dimensions, and nine plus or positive sizes. In one set of eighteen, the + and − blocks have a range of 0·0001 to 0·0009 in., in the other set the range is + and − from 0·001 to 0·009 in.

Thus, required—

4 − 0·005 in.	6 + 0·0065 in.
0·1 − 0·005 ,,	0·1 + 0·0005 ,,
0·9 ,,	0·1 + 0·006 ,,
3 ,,	0·8 ,,
	5 ,,
4 − 0·005 ,,	6 + 0·0065 ,,

There are several accessories by the use of which the utilities of the blocks are facilitated and extended. A set of large blocks is made up to 20 in. by which the range of the fractional blocks is much extended. Holders are provided for the blocks by which jaws are set (Fig. 15) for gauging external surfaces, and for bores. Points are set for checking graduations (Fig. 16). Scriber points are set from a base block (Fig. 17). Adjustable snap gauges (Fig. 18) are set with the blocks, and adjustable plug gauges similarly.

The Johansson blocks are unique, and their manufacture is secret. Their remarkable feature is, that when the surfaces are wiped, and the blocks are slid over one another to expel air, they will adhere like magnetized steel. Though turned about in any positions they will not separate. This must be due to more than atmospheric pressure, probably to molecular attraction. The surfaces are highly finished like burnished silver, and the thicknesses are so perfectly parallel that turning about the blocks end for end does not affect the accuracy of the combination.

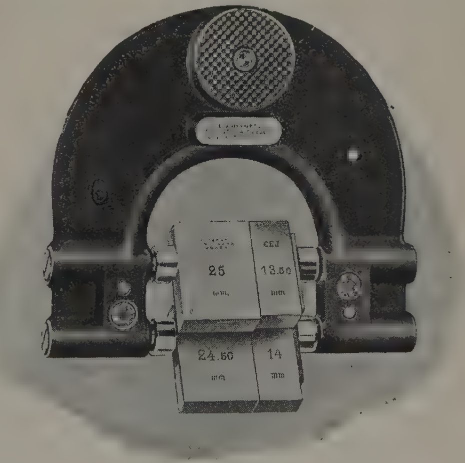


FIG. 18. SETTING A SNAP GAUGE WITH JOHANSSON BLOCKS

Micrometer Calipers. To measure very fine dimensions that are not fixed in gauges, the micrometer principle is embodied in calipers, external and internal, and in some other instruments. The principle is simple, but the mechanism is delicate, and is varied in the hands of different manufacturers. An accurate screw of fine pitch is moved longitudinally through a nut, and fractional parts of the pitch are obtained by turning the nut through a portion of a circle. One complete turn of the nut will advance the screw to a distance equal to one pitch, a quarter turn to a quarter of the pitch, and so on. A piece of work set between an extension of the screw and a fixed anvil will be measured minutely and precisely.

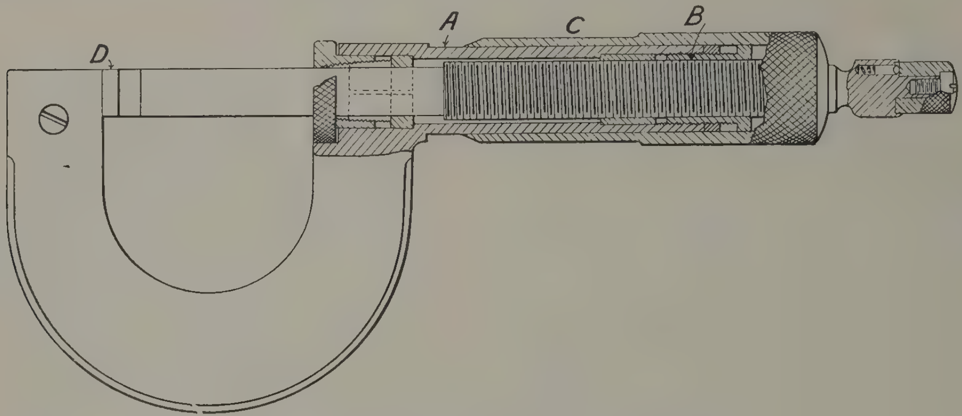


Fig. 19. Section through Micrometer Caliper

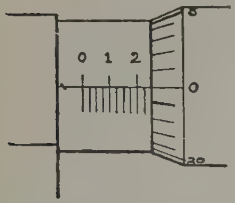


Fig. 20.
A Micrometer
Reading

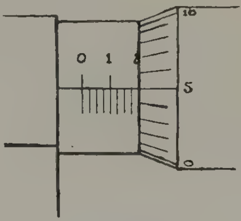


Fig. 21.
A Micrometer
Reading

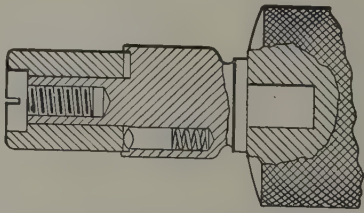


Fig. 22. Ratchet Stop and Spring Pawl to Prevent Excess of Pressure on Micrometer Screw

Fig. 19 gives a section through a Brown & Sharpe caliper. The horseshoe frame is extended to form the sleeve or barrel *A* which encloses the screw *B*. Another sleeve *C*, the thimble, surrounds the barrel. It is coned at the end nearest the horseshoe, and is graduated around to meet a zero line on the barrel. When the spindle makes contact with the anvil *D* as in the figure, the zero line on the barrel and that on the graduated thimble correspond. Provisions are made as seen for protecting the screw and for taking up wear, and in some cases for locking after a dimension has been set.

In micrometers graduated for English dimensions the screw has a thread of 40 to the inch. The barrel is crossed with graduations in front of the thimble, with the pitch distances of every fourth division being marked. As one division equals $\frac{1}{40}$ in. it is convenient to reckon decimally, as 0 in., 0·1 in., 0·2 in., 0·3 in., etc. There are 25 subdivisions round the thimble. To take a reading, the fortieths visible on the barrel are noted, then the tenths, and then the thousandths on the bevelled edge of the thimble. As there are 25 subdivisions round the thimble, and one-twenty-fifth of its revolution moves the screw along $\frac{1}{25}$ of $\frac{1}{40}$ in, this equals $\frac{1}{1000}$ in. In Fig. 20 the zero lines correspond. In Fig. 20 the reading on the sleeve is $\frac{2}{10} = 0{\cdot}2$ in., and $\frac{2}{40} = 0{\cdot}05$ in., totalling to 0·25 in. which is the measurement, since the zero lines correspond. In Fig. 21 the reading is $\frac{9}{10}$ = 0·2 in. But the thimble is moved round to the fifth division, equal to $\frac{5}{1000}$ = 0·005 in. which has to be added to the $\frac{9}{10}$, making the measurement 0·205 in. It is easy to subdivide the divisions with the eye, for half- and quarter-thousandths. The capacities of these instruments mostly range from 1 to 2 in. Larger sizes are made in different frames, covered with rubber to prevent expansion by the heat of the hand. Tail spindles or anvils are fitted interchangeably in these.

It is easy to vitiate the readings by exercising too much pressure on the screw. To prevent this a ratchet stop is often fitted with a spring pawl (Fig. 22). Excess of pressure causes the ratchet to slip past the pawl and arrest the further turning of the measuring spindle.

When positive measurements finer than thousandths are required, the vernier is combined with the micrometer. The micrometer is fitted to inside calipers, to depth gauges, to beam calipers, to the measurement of screw threads, and all to English and metric dimensions.

NOTABLE EVENTS IN ENGINEERING HISTORY

By E. A. Forward, A.R.C.S., M.I.Mech.E.

LESSON IX

INDUSTRIAL ENGINEERING

In the production of the necessities of life, the engineer has a wide field of operation, and the great advances which took place during the late eighteenth and early nineteenth centuries were largely due to the application of steam power. The engineer has thus been largely responsible for the increased production, with reduction of cost, which has rendered the various commodities available to the masses of the population.

In early times the manufacture of necessities, such as food and clothing, was carried on in the homes of the people, who made what was required for their own families with simple tools or hand-operated machines, but the use of machinery of greater capacity, with various forms of power to drive it, led to the concentration of such work in factories.

Agriculture. Although agriculture is the oldest and most important industry, but little attention was paid to the improvement of the implements used, and it is only since about 1850 that this large field for the use of mechanical appliances has received serious attention.

The plough was the only agricultural instrument used by the Ancients, and amongst primitive tribes still remains in its early form of a forked bough with one branch cut short and pointed. The Romans added the coulter, or front-cutting knife, and the mould board which turns over the furrow slice. The more extensive use of iron followed, but the greatest single improvement made in modern times was the chilled cast-iron ploughshare invented by Robert Ransome in 1803. Ploughing by steam power was tried as early as 1832, but it did not become a practical proposition until 1855. Machines for sowing seed appeared during the first half of the nineteenth century.

A form of reaping machine is said to have been used in Gaul about A.D. 23, but little advance in this direction was made until early in the nineteenth century, when James Smith invented a machine with a revolving disc cutter. The first efficient reaping machine on the shearing principle was that made by the Rev. Patrick Bell in 1826 (Fig. 35). This was followed in 1831 by McCormick and Hussey's machine in America, but such machines did not come into general use until after the Great

Exhibition of 1851. These machines were subsequently made to bind the sheaves.

In 1787, Andrew Meikle produced a threshing machine which embodied most of the essentials of the present-day machines, although in a very imperfect form. The latest machines perform all the operations of separating the grain from the straw, chaff, broken ears, dirt, and stray seeds, and they grade it into different qualities.

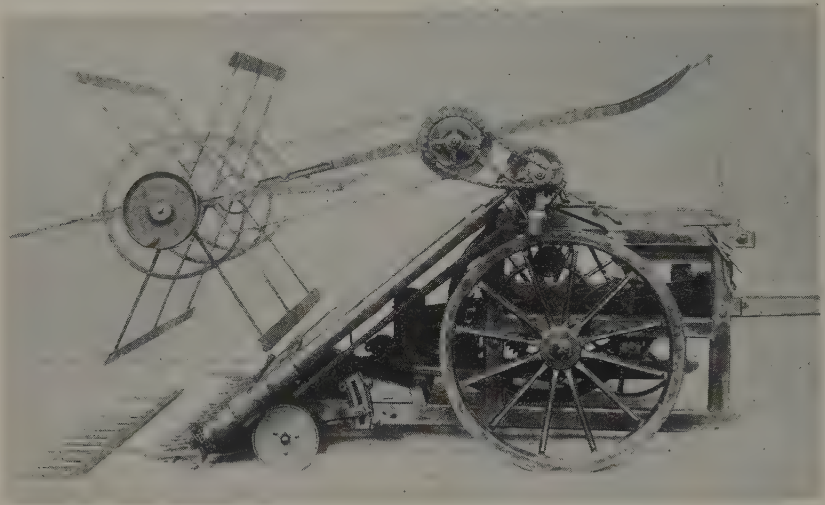


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FIG. 35. BELL'S REAPER, 1826

Flour Mills. The earliest method of reducing corn to flour was with a pestle and mortar. The hand quern succeeded these and acted like our present millstones by a circular grinding action. The stones increased in size and were then driven by water (Fig. 36) or wind, and finally by steam power. The meal from the stones was dressed and sifted through screens to remove the bran.

The gradual reduction process, in which the corn is reduced to flour in successive stages and the product collected from each stage, originated in Hungary, and led to the introduction of metal rollers in place of millstones. Roller milling had been tried at Paris in 1820, but was not successfully adopted until 1867 at Budapest. The introduction of this system into England was slow, the first complete and automatic gradual reduction mill being erected at Bilston in 1879.

Textiles. The earliest appliance for twisting fibres into a thread appears to have been the distaff. The spinning wheel which followed was introduced into this country in the sixteenth century and continued in use for a long period.

One of the first steps in the development of cotton spinning was the machine invented by James Hargreaves, between 1750 and 1757, the principal merit of which lay in the fact that one person could spin 120 threads at a time, instead of only one as with the spinning wheel. The method of drawing out a roving finer by passing it between pairs of rollers running at different speeds was invented by Lewis Paul in 1738, but

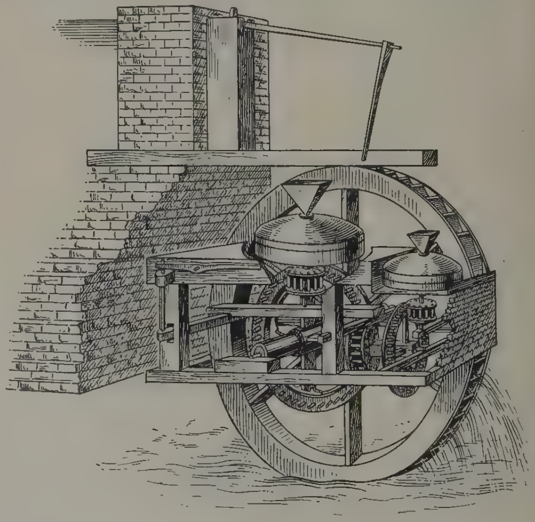

FIG. 36. CORN MILL, 1723

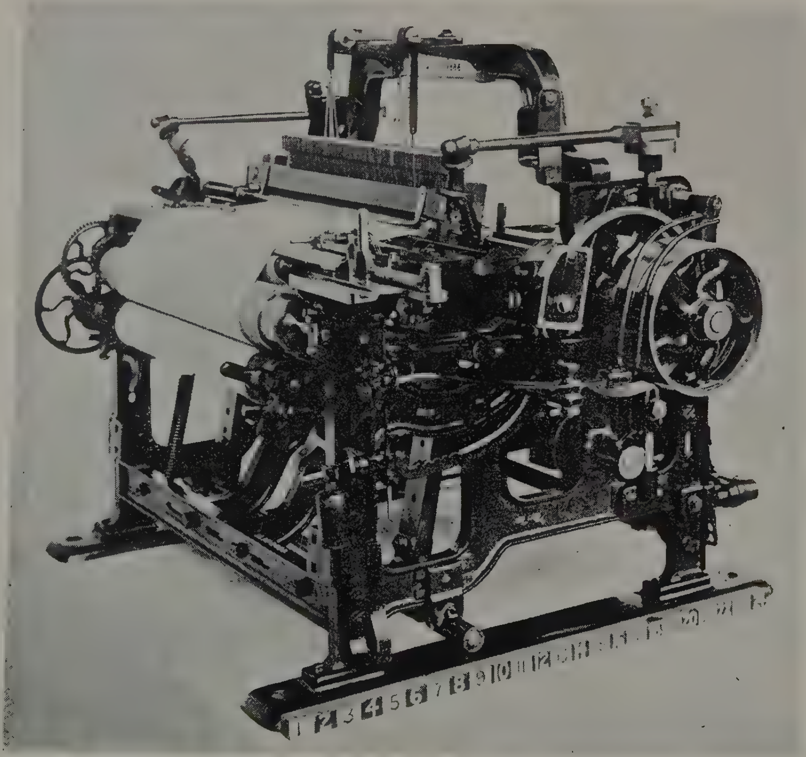

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FIG. 38. POWER LOOM

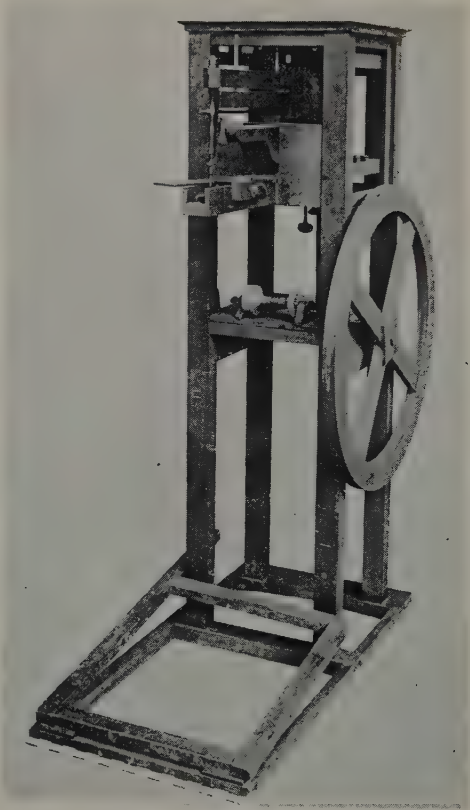

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FIG. 39. THIMMONIER'S SEWING MACHINE, 1830

was not successfully applied until Sir Richard Arkwright used it in his first spinning machine of 1769 (Fig. 37). The "mule" spinning frame, the only form with which very fine yarns can be spun, was invented by Samuel Crompton in 1774. He and the other great inventors of the time suffered severely from poverty and mob violence, the one exception being Arkwright who succeeded in making his machinery commercially successful. The mule was made self-acting by Richard Roberts in 1825. The ring-spinning frame was invented in America about 1828, and although unable to spin the finest yarns, is of the greatest importance from its simplicity and large output.

Weaving is of earlier origin than spinning, but little improvement was made in the primitive hand looms until early in the eighteenth century. In 1733, John Kay invented the fly shuttle by which the speed of weaving was greatly increased and the labour diminished, while in 1760, Robert Kay invented the drop box by which several shuttles with different coloured wefts could be used.

The power loom (Fig. 38) was introduced by the Rev. Edmund Cartwright in 1785, a steam-engine being used in his factory in 1789.

The Jacquard apparatus for producing the most intricate patterns was invented partly by Falcon in 1728, partly by Vaucanson in 1745, and completed by Jacquard in 1804. In recent years automatic looms which can change their own weft bobbins have been developed, such as the Northrop and Steinen looms. These are arranged to stop working if any fault occurs.

Knitting is supposed to have originated in Scotland in the fifteenth century, and the first machine for doing this work was the stocking frame invented by the Rev. William Lee in 1589. Lee failed to obtain support in England and took his machines to France, but in 1640 his frames were adopted at Leicester. A machine for producing ribbed knitting was made by Jedediah Strutt in 1758, while a practical circular knitting machine was devised by Peter Claussen in 1845, although earlier attempts had been made. In 1858 the now universal latch needle was introduced by Mr. M. Townsend.

Sewing. Bone needles were known in prehistoric times and the common steel needle was probably introduced into England in the sixteenth century. Attempts to produce a sewing machine date from 1790, but the first practical machine, making a chain stitch, was invented by B. Thimmonier in 1830 (Fig. 39). The first successful lock-stitch machine was that produced by Elias Howe in 1845, while I. M. Singer, who patented his first machine in 1851, did much to accelerate the commercial introduction of the sewing machine.

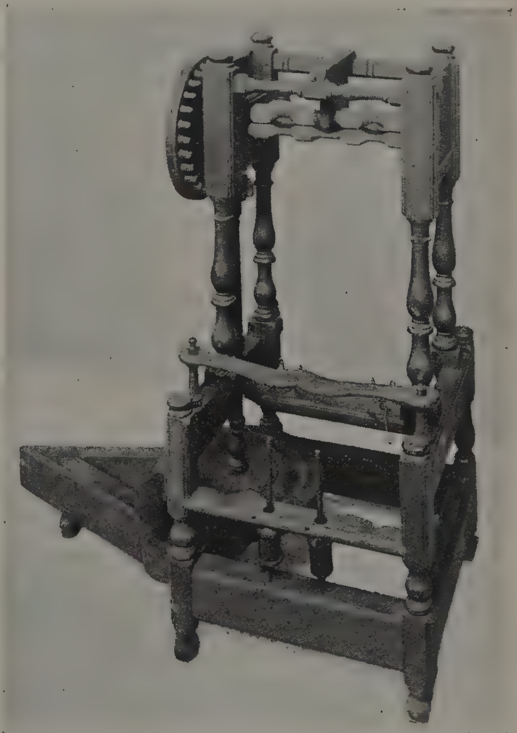

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FIG. 37. ARKWRIGHT'S SPINNING MACHINE, 1769

PUMPS: CONSTRUCTION AND MAINTENANCE

By Owen A. Price, M.I.Mech.E.

LESSON VIII

HYDRAULICS OF THE CENTRIFUGAL PUMP

Now that some acquaintance has been made with practical constructions it will be convenient to consider the hydraulic design of the centrifugal pump.

In the first place we shall review some of the phenomena relating to fluids in motion. Rigorous treatments of fluid motion can be found elsewhere, our object is to present the pump engineer's conception of these matters without which a proper understanding of the centrifugal pump is impossible.

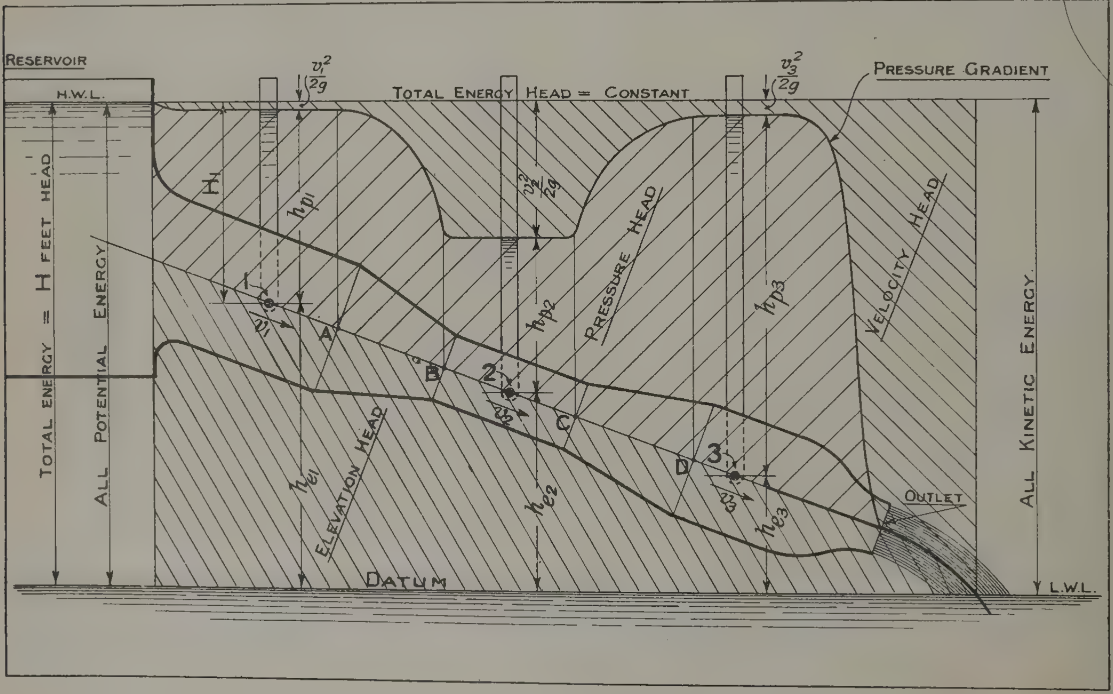


Fig. 29. Bernouilli's Theorem—Geometrical Representation

Equation of Continuity. In connection with flow through a water passage it is an axiom, employing proper units, that

Quantity = velocity × area = constant

This is termed the *equation of continuity*, and applies to the *mean velocity* when the cross-sectional *area* in question is measured perpendicular to the direction of the mean velocity. From this relation it follows that in a *tube of flow* (such as that shown later in Fig. 29) the velocity is inversely proportional to the area.

A *break in continuity* implies, either that the fluid stream has parted asunder, or a surface of discontinuity has been formed, as, for instance, when a stream rends its way through surrounding walls of turbulent eddies, or where *foaming* is produced.

Bernouilli's Theorem. Applying the principle of the *conservation of energy* to the flow of a fluid through a passage of gradually varying section, we obtain the relation between the potential and kinetic energies for an unchanging total energy.

Thus, considering flow at the three points 1, 2, and 3, in a supposed frictionless tube (Fig. 29), with velocities v_1, v_2, and v_3 ft. per second, pressure heads h_{p1}, h_{p2}, and h_{p3} ft., and elevation heads h_{e1}, h_{e2}, and h_{e3} ft. above a reference datum, we have the *total energy* (H ft. head), always equal to the sum of the *pressure, elevation,* and *velocity* heads, that is—

$$H = h_{p1} + h_{e1} + \frac{v_1^2}{2g} = h_{p2} + h_{e2} + \frac{v_2^2}{2g}$$

$$= h_{p3} + h_{e3} + \frac{v_3^2}{2g} = \text{constant}$$

This relation holds good for all points in the system and is known as *Bernouilli's Equation.*

If the axis of the tube is horizontal through the point 1, the expression simplifies to—

$$H_1 = h_{p1} + \frac{v_1^2}{2g} = h^1{}_{p2} + \frac{v_2^2}{2g}$$

$$= h^1{}_{p3} + \frac{v_2^3}{2g} = \text{constant}$$

Bernouilli's equation is the standard of reference for the *ideal* case in hydraulic flow, and is of fundamental importance. Fig. 29 forms a geometrical representation of the theorem which also may be expressed in the following words: When water flows through a passage of gradually varying section so that energy is not wasted and the total energy head therefore remains unaltered, there is an interchange of pressure head with velocity head, these quantities being so related that their sum remains constant.

Conversions of Energy. The nature of the fluid motion during a transformation of *pressure energy into velocity energy* differs radically from the sort of motion occurring with a transformation of *velocity energy into pressure energy*. That is to say, there is an essential difference between flow along a gradually contracting passage (A to B in Fig. 29), and flow along a gradually expanding passage (C to D in Fig. 29).

It is common knowledge that, in an unconstrained fluid, flow will take place naturally from a region of high pressure to a region of low pressure; in fact, the existence of a pressure difference constitutes a *velocity potential*. Flow, in this case, is the natural process of restoring equilibrium, and the resulting stream, if unaffected by other forces, is *stable*, or is a *steady* movement in conformity with the natural tendency.

With this fundamental notion in mind we shall examine the particular case of flow in circular section taper pipes.

Convergent Flow. With *convergent flow* (Fig. 30, diagram A) the liquid travels in the same direction as the *velocity potential,* that is, it flows from a relatively *high* pressure region into a relatively *low* pressure region. The distribution of axial velocity across the diameter of a section is shown both at entrance to, and exit from, the convergent pipe, and the *lines of motion* are seen to close in towards one another and equalize in velocity, so that the distribution is much more nearly uniform at exit than at entrance. The lines of motion are not subject to any contrary influence which might act as a source of disturbance, and the flow is *stable.* Such flow conforms closely to Bernouilli's equation. Using the notation of Fig. 29 and assuming a horizontal axis, the fall of pressure, therefore, is practically $\frac{v_2^2 - v_1^2}{2g}$ ft. head.

Divergent Flow. With *divergent flow* (diagram B) the liquid travels in a direction contrary to the *velocity potential*, that is, it flows from a relatively *low* pressure region into a relatively *high* pressure region. The stream, as it enters the divergent pipe, continues its straight line direction, thus tending to leave a vacant annulus, which, on account of the contrary *velocity potential*, is filled by backward turning eddies ravelled off the outer filaments of the main stream. Thus, the *velocity potential* (in this case a higher downstream pressure) constitutes a contrary influence acting as a second source of motion, and forming a turbulent region which operates as a drag on the whole body of water. As the lines of motion are retarded they splay out, or *diverge* from the axis, in the manner shown, expending a portion of their kinetic energy in eddying and wall friction, whilst the remainder is converted into pressure energy. The diagram showing the velocity distribution at exit, indicates that to a considerable extent the centre stream ploughs its way through a trailing wake of more slowly travelling water. This disturbed kind of flow is *unstable*, or *unsteady* motion, and results unavoidably in dissipation of energy.

When we realize that the backward turning eddies cause the destruction of energy, it is apparent that the losses will be greater when the walls diverge at wide angles than at small angles, because, within limits, the facilities for formation of these eddies increase as the angle of divergence increases.

Using the notation of Fig. 29, the ideal rise of pressure, according to Bernouilli's equation, is given by $\frac{v_2^2 - v_3^2}{2g}$ ft. head.

The actual rise, however, will be $k\left(\frac{v_2^2 - v_3^2}{2g}\right)$ ft. head, only, where k is less than unity, and varies with the roughness of the walls as well as with the angle of divergence. For the range of angles likely to be used in centrifugal pump work, k has the following approximate mean values for circular pipes—

Angle of Divergence	k
6° to 10°	0·87
10° to 15°	0·87 to 0·82
15° to 20°	0·82 to 0·73
20° to 25°	0·73 to 0·60

Discussion of divergent flow in rectangular or other passages would involve consideration of an infinite variety of proportions, the above figures, however, will serve as an approximate guide for all expanding passages.

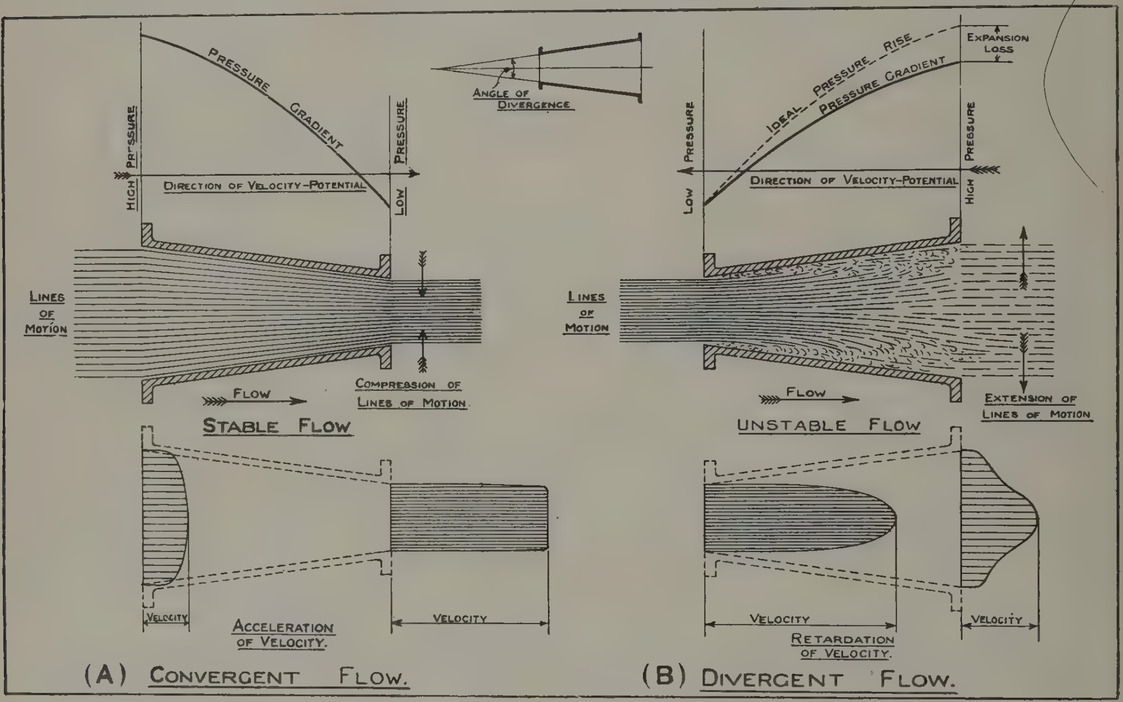


FIG. 30. GRADUAL CHANGES OF STREAM SECTION

At the outlet ends of the two taper pipes in Fig. 30, arrows are drawn, transversely to the stream, suggesting the ideas of compression and tension. It is helpful to consider a *contracting* stream as *in compression* under the reaction from the convergent walls, and an *expanding* stream as *in tension*, due to the divergent spreading of the lines of motion. The *compression* of a liquid suggests firmness or *stability*; also, it is known that liquid cannot sustain more than an infinitesimal amount of tension without rupture, and, therefore, *tension* suggests looseness or *instability* conducing to rupture of the lines of motion with consequent eddy formation.

Sudden Enlargements. With an abrupt *expansion* of passage area there is a sudden release of pent-up energy, and the stream proceeds at a lower velocity. The result is, the whole of the energy represented by the difference in velocity is lost because there is no provision for converting this *velocity-difference* into pressure.

Therefore, with an abrupt reduction of

velocity from V to v ft. per second (see Fig. 31, *A*), the entire head due to the relative velocity $(V-v)$ is lost, hence

$$\text{Loss} = \frac{(V-v)^2}{2g} \text{ ft. head.}$$

The ideal rise of pressure head would be, by Bernouilli's equation,

$$\text{Ideal rise} = \frac{V^2 - v^2}{2g} \text{ ft. head.}$$

Therefore the actual pressure rise is

$$\text{Actual rise} = \frac{V^2 - v^2}{2g} - \frac{(V-v)^2}{2g} = \frac{v(V-v)}{g} \text{ ft. head.}$$

When dealing with sudden changes of velocity it is often desired to estimate losses quickly, and to note at once the effect of altering the amount of *velocity-change.* For this purpose it is convenient to express the loss in terms of the velocity head possessed by the moving water, in the manner indicated in Lesson II.

If a and A are the areas respectively of the small and large passages (Fig. 31, *A*), it is evident that

$$v = V \times \frac{a}{A} \text{ and } V = v \times \frac{A}{a}$$

Therefore the loss, given above, for the sudden enlargement, may be expressed in terms of the passage areas by

Loss of head in ft.

$$= \left(1 - \frac{a}{A}\right)^2 \frac{V^2}{2g}, \text{ or } \left(\frac{A}{a} - 1\right)^2 \frac{v^2}{2g}$$

$$= k\frac{V^2}{2g}, \text{ or } K\frac{v^2}{2g}$$

according as the loss is related to V or v.

The coefficients k and K are easily calculated for any *ratio of enlargement* $\frac{A}{a}$, and may be tabulated as follows—

$\frac{a}{A}$	k	K
0·05	0·9025	361·0
0·1	0·81	81·0
0·2	0·64	16·0
0·3	0·49	5·44
0·4	0·36	2·25
0·5	0·25	1·00
0·6	0·16	0·44
0·7	0·09	0·185
0·8	0·04	0·0625
0·9	0·01	0·012

Sudden Reductions. When an abrupt *reduction* of passage area occurs, the stream suffers a neck contraction (Fig. 31, *B*) at the inlet to the reduced passage, and thereafter expands to fill the smaller passage.

There is practically no energy loss in the convergent portion of the flow, but below the constricted neck a *sudden enlargement loss* occurs, the amount of which is dependent upon the contraction. On the basis of a *sharp edge* at the juncture of the passages, various rules have been proposed for relating the *coefficient of contraction* (and therefore the sudden enlargement loss) to the ratio of passage reduction. In pump design, however, we carefully avoid sharp edges, and are more concerned with the contractions and consequent losses when the edge is rounded.

The following table shows the effect of rounded edges (Fig. 31, *B*) in reducing the contraction and, therefore, reducing the head loss—

Ratio $\frac{r}{d} = \frac{\text{Edge Radius}}{\text{Diam. of Reduced Section}}$	Coefficient of Contraction $= \frac{\text{Neck Area}}{\text{Reduced Area}}$	Loss Coefficient K in formula— $\text{Loss} = K\frac{v^2}{2g}$ ft.-head
0·00 (sharp edge)	0·64 (about)	0·5 to 0·32
0·05	0·71	0·16
0·10	0·77	0·09
0·20	0·86	0·027
0·30	0·93	0·005
0·36 (or larger)	1·00	0·0

In the above table the loss (expressed in terms of the reduced passage velocity v in ft. per second) has been calculated by the sudden enlargement method previously discussed. The higher velocity V in the earlier discussion corresponds with the velocity at the neck contraction and, as the *coefficient of contraction* is equivalent to the *ratio of enlargement,* the loss coefficient K for corresponding contractions is identical with K in the earlier table. The figures are approximate only, but they serve to emphasize the very great value of rounded corners at entrances to passages, or wherever reductions of passage area occur.

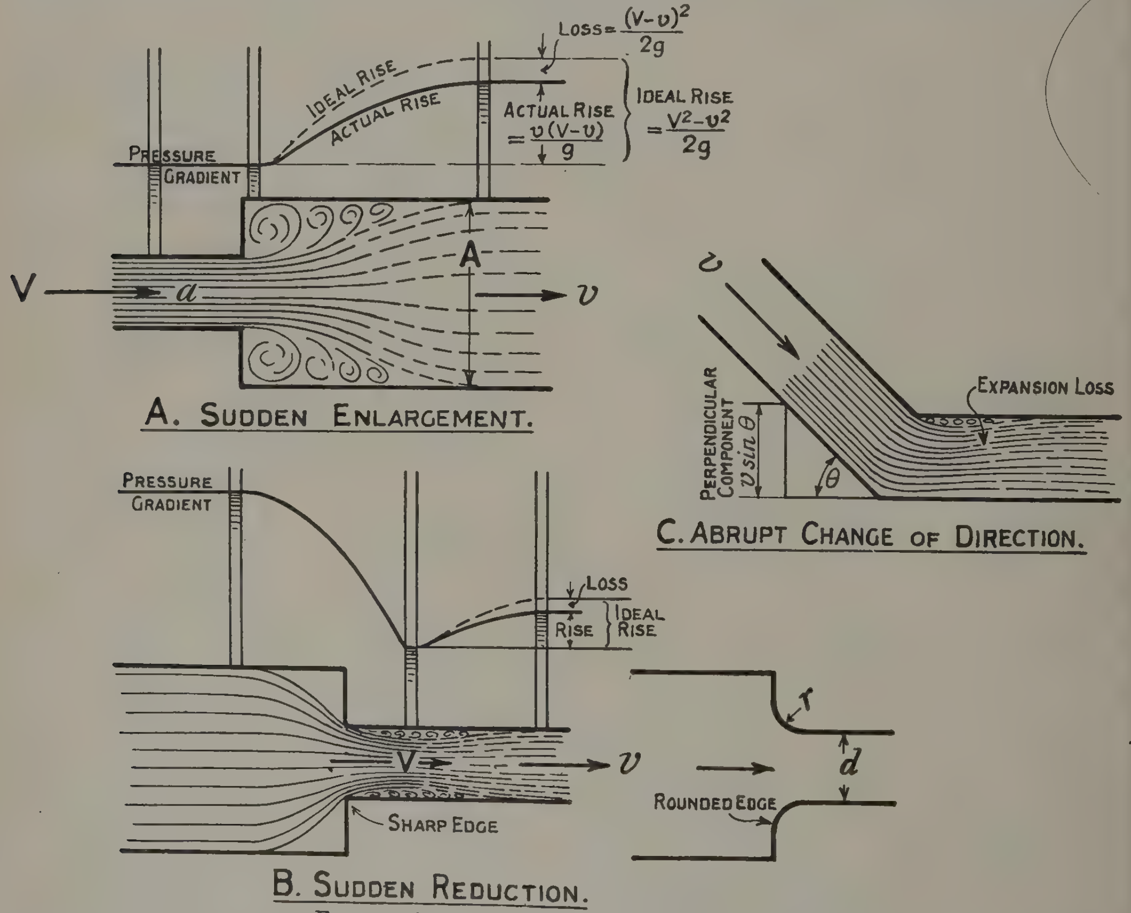
Abrupt Change of Direction. A twofold loss occurs at an abrupt change of *direction* (Fig. 31, *C*). The energy in the velocity component ($v \sin \theta$ ft. per second) perpendicular to the new direction of flow, is wasted in *shock,* and, in addition, there is a loss due to the re-expansion of the jet after contraction at the abrupt corner. Hence

$$\text{Total loss of head in ft.} = \frac{(v \sin \theta)^2}{2g} + \text{the}$$

expansion loss.

After determining the contracted area, the expansion loss is treated in the manner previously indicated.

Various small scale experiments have indicated total losses considerably smaller than would be given by the above method, and the inference is that when a stream is diverted *obliquely* the change of direction is *not abrupt*, but follows a somewhat curved path.


Loss = $\frac{(V-v)^2}{2g}$
Ideal Rise
Actual Rise
Pressure Gradient
Actual Rise = $\frac{v(V-v)}{g}$
Ideal Rise = $\frac{V^2-v^2}{2g}$
A
V
a
v
v
Expansion Loss
Perpendicular Component $v \sin \theta$
θ
Pressure Gradient
Loss
Rise
Ideal Rise
r
d
V
v
Rounded Edge
Sharp Edge

A. Sudden Enlargement.

C. Abrupt Change of Direction.

B. Sudden Reduction.

Fig. 31. Abrupt Changes of Stream Section

Curved Streams. In the previous discussions the lines of motion have been, for the most part, symmetrically disposed about a straight-line axis. Under these conditions the pressure distribution, transverse to the stream, is practically *uniform* across any particular horizontal plane, and *varies* hydrostatically across any particular vertical plane.

If, however, the water is travelling along a curved path, every particle has an acceleration towards the centre of curvature, in conformity with the ordinary laws of circular motion. Thus, if O in Fig. 32, is the centre of curvature, r ft. the radius of the curved line of motion, and w ft. per second the circumferential *velocity of whirl*, then every particle on the curved path will have an acceleration $\frac{\omega^2}{r}$ ft. per second per second towards O. The head h required to impart this centripetal acceleration and to produce the *deviating force* ρh (ρ = density), necessary to bend the line of motion, is derived by reaction from the walls, or by reaction from a surrounding body of water at suitable pressure.

The *resistance* which the deviating *force* has to overcome is the *centrifugal force* of the water, termed *centrifugal pressure* in hydraulics, acting outwards along the radius of curvature. As this force, or pressure, is a function of the radius it is evident that in a curved stream the pressure is *not uniform* transverse to the stream, along the radius of curvature, but increases with the radius.

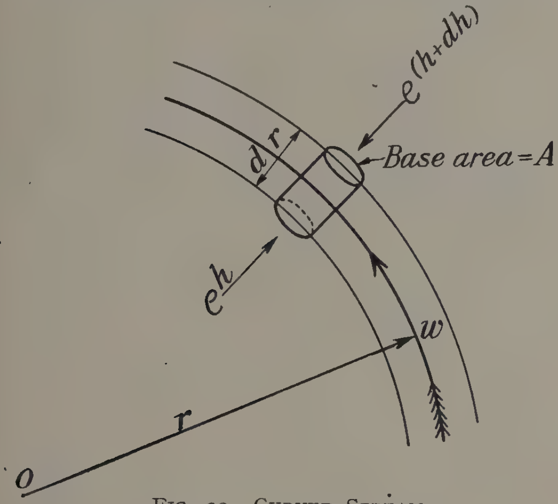


FIG. 32. CURVED STREAM

Consider the forces acting on an indefinitely small fluid cylinder of length *dr*, base area *A*, and weight $\rho.dr.A$ (ρ= density). The centrifugal force acting outwards is

$$\rho\,\frac{.\,dr\,.\,A}{g} \times \frac{\omega^2}{r}$$

which is balanced by the net radial pressure on the ends of the cylinder, that is, $\rho h\,A$ on the inner end and $\rho(h + dh)\,A$ on the outer end, or $\rho\{h + dh) - h\}\,A = \rho.dh.\,A$ net radial inward pressure.

$$\therefore\ \rho.\,dh\,.\,A = \frac{\rho\,.\,dr\,.\,A}{g}\,.\,\frac{\omega^2}{r}$$

$$\text{whence } \frac{dh}{dr} = \frac{\omega^2}{gr}$$

which is the general relationship for variation of head, along the radius of curvature, with change of radius. When the variation of the whirling velocity ω with the radius r, is known, we can, by integration, determine the change of head h along the radius.

FUEL

By ARTHUR GROUNDS, B.Sc., A.I.C., Assoc.M.I.Min.E., A.M.C.T.

LESSON IX

SPECIAL FUELS AND FURNACES

WE have so far considered mainly the combustion of coal as a fuel, but we must briefly study the special precautions which must be observed when other fuels have to be burnt. The theoretical considerations outlined in the first chapters apply equally, of course, to all fuels, and provided that we make an analysis of the fuel to be burnt, we can estimate what will be the composition of the flue gas with varying amounts of excess air, and also what losses are likely to occur at definite temperatures. In many tropical countries, wood is the main source of fuel, whilst sugar cane refuse (known as bagasse or megasse), rice husks, nut shells, peat, rotting grain, and other strange fuels are often burnt in considerable quantities. In order to give the reader some idea of the difficulties of burning such fuels, it will be advisable first of all to set down in tabular form the composition of such combustible materials as are generally met with, in order to show how there is a gradual and progressive change from pure cellulose, which is the basis of wood, to anthracite, which is the natural fuel which has undergone the most severe changes. In the table overleaf the analyses are given of these fuels on a basis which eliminates moisture, ash, and sulphur.

It must be realized that the given analyses are only representative of "types" of fuels, and must not be taken to apply to all fuels of that particular class. For example, not all woods have the composition given, but the analyses shown will serve as a guide to the approximate composition. In burning wood and bulky fuels of a like nature, it is essential to provide plenty of combustion space, and for this reason, a "stepped" grate is often provided, this being a grate in the form of a terrace, down which the fuel is fed so that at the top of the grate water is driven off from the fuel, whilst as it slowly descends, carbon dioxide and other inert gases are evolved until, when the lower sections of the grate are reached, the fuel is in the form of charcoal. Where stepped grates are not provided, successful combustion can frequently be secured by dropping the usual grate below the normal level, as, for example, in the flues of a Lancashire boiler. A considerable quantity of wood is burnt even in this country, consisting of refuse from sawmills, woodworking shops, furniture factories, etc. The heating value of dry wood ranges from 7,300 to 9,900 B.Th.U.'s per lb., whilst the moisture content varies from 20 to 60 per cent, with an average of about 45 per cent. Bagasse is burnt very largely on the sugar fields of America and the East Indies, and consists of fibre, various sugars, and water. The fibre content forms from 50 to 60 per cent of the weight, sugars form about 10 per cent, and the water forms from 30 to 50 per cent. In the raw state, bagasse has a calorific value of from 3,600 to 4,800 B.Th.U.'s per lb., but the vegetable refuse is sometimes wholly or partially dried before combustion. Peat is sometimes burnt in briquette form, after suitable treatment, consisting of drying and, occasionally, carbonization, whilst considerable success has been attained in plants in which peat has been burnt in powdered form, since this fuel lends itself readily to pulverization and burns freely, being easily ignitable.

Substance	Carbon	Oxygen	Hydrogen	Nitrogen
Pure cellulose . .	44·5	49·4	6·1	—
Wood . . .	50·0	43·0	5·7	1·3
Peat . . .	60·0	32·5	5·5	2·0
Lignite . . .	66·5	26·5	5·5	1·5
Sub-bituminous coal	75·0	18·0	5·5	1·5
Bituminous coal .	82·0	11·0	5·4	1·6
Semi-bituminous coal	86·5	7·0	5·0	1·5
Semi-anthracite coal	91·0	4·0	3·5	1·5
Anthracite . .	93·2	3·0	2·5	1·3
Graphite anthracite	97·3	2·1	0·5	0·1

Liquid fuels are of immense importance, and it is unfortunately impossible in this particular series of lessons to even briefly touch on the application of oil fuel. Natural oils, tar, gas oil, paraffin, petrol, are all of great importance in the field of mechanical engineering, whether considered as fuels for steam raising or for use in internal combustion engines. The heavy fuel oils are usually burnt in steam raising plants in burners provided with either steam or compressed air-operated jets so that the liquid fuel is atomized into a fine spray after heating, air being mixed thoroughly with the fuel by vanes, either stationary or whirling. Increased combustion space is required for the most satisfactory operation of liquid fuel-fired plants. Next to giving details of the various fuels available, the best advice the writer can give to his readers is to study carefully one or more of the following books, in which full particulars will be found of the composition, properties, and methods of utilization of the many various types of combustible—

Coal and its Scientific Uses, Bone (Longmans, Green).

Coal—Its Origin, Method of Working, and Preparation for the Market, F. H. Wilson (Pitman).

Boiler Inspection and Maintenance, R. Clayton (Pitman).

Mechanical Stoking, Brownlie (Pitman).

Fuel Economy in Steam Plants, Grounds (Pitman).

Oil Power, S. H. North (Pitman).

Patent, Smokeless and Semi-smokeless Fuels, J. A. Greene and F. M. Perkin (Pitman).

Liquid Fuels for Internal Combustion Engines, H. Moore (Chapman & Hall).

Coal Carbonization, J. Roberts (Pitman).

Factory Wastes as Fuels, A. Reid (Benn Bros.).

The Practical Chemistry of Coal and Its Products, A. E. Findlay and R. Wigginton (Benn Bros.).

Theory and Practice of Combustion, J. E. Lister and C. H. Harris (Benn Bros.).

Mechanical Draught, J. E. Lister and C. H. Harris (Benn Bros.).

Steam Power Plant Engineering, G. F. Gebhardt (John Wiley & Sons).

The Calorific Value of Fuel, Coste and Andrews (Griffin).

Handbook of Briquetting, Francke and Lantsbury (Griffin).

Read also the following periodicals, which give particulars of research work and valuable contributions on the subject of fuels in general—

Fuel in Science and Practice.
The Fuel Economist.
Combustion.
The Colliery Guardian.
The Iron and Coal Trades Review.
Oil Engineering and Technology.
The Journal of the Society of Chemical Industry.
The Journal of the Institute of Petroleum Technologists.
The Gas World.
The Gas Journal.

(CONCLUSION)

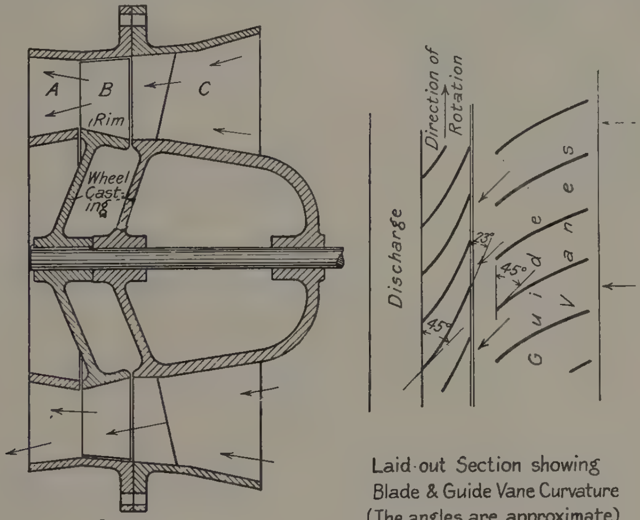
FANS AND BLOWERS
THEIR CONSTRUCTION AND OPERATION

By F. G. Whipp, A.M.I.Mech.E., A.M.I.H.V.E.

LESSON VI

PRESSURE WITH SCREW FANS

It was stated in the beginning of these lessons that under some conditions very slight pressures were possible with propeller fans, although this meant a reduction in efficiency. Attempts have been and are still being made to adapt the propeller principle to the creation of pressures on efficient lines, both by certain adjustments to the fan blades and by introducing several stages of propellers.



Section

Laid-out Section showing Blade & Guide Vane Curvature (The angles are approximate)

Fig. 17

Where, however, large volumes of air have to be handled, mechanical considerations make for a costly apparatus if this method is adopted, especially when compared with the economically constructed *centrifugal fan* of present-day design.

Immediately resistance is offered to the free flow of air of a propeller fan, air will re-enter the wheel at a point where the velocity is lowest which, as we have already seen, is near the centre.

This re-entry can be prevented by fitting a disc over the central portion of the wheel, the disc being of such a diameter as to extend to that radius of the impeller where the velocity commences to reach a maximum figure.

Such an experiment was described by Mr. W. G. Walker in a paper previously cited, and he found that a better efficiency was obtained with this combination when a resistance was present, than when working in "free air." This obviously is to be expected and proves the soundness of policy when reckoning the area of the central boss as a positive quantity in all calculations.

The tests also established the fact that better conditions prevail when the fan is exhausting from a restricted space, than when delivering into such a space, a state of affairs to be expected in view of the fact that a maximum air velocity is not attained until after the column of air has traversed the fan wheel, and since pressure varies as the square of the velocity.

Attention has been drawn to this experiment in order that the reader may the better understand the principle of the Rateau screw fan, which bears out these characteristics. This fan was designed by Professor Rateau, who increased the pressure efficiency by introducing guide vanes and a casing.

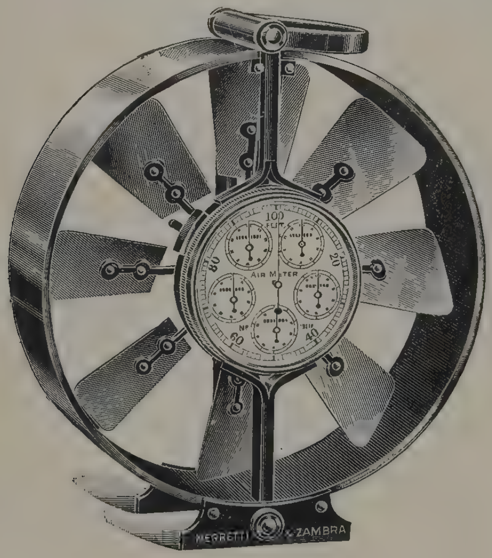
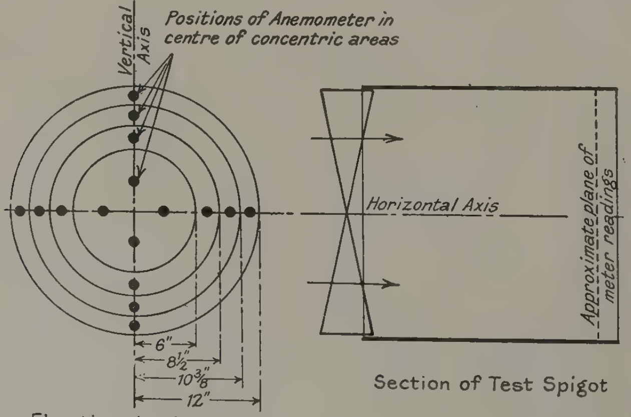


FIG. 18. AIR METER

The casing takes the form of an annular inlet chamber in which the guide vanes are housed, see *C*, Fig. 17. The internal wall of this chamber is stream-lined, whilst the external portion tapers towards the impeller *B*. The discharge casing *A* is really an expanding mouthpiece, and assists in the conservation of energy by converting kinetic energy into static pressure. This mouthpiece is sometimes extended with a further saving of energy. The whole system is so designed that the losses through restriction of orifice are reduced to a minimum, as the form of passage through which the air stream passes conforms to the natural flow of air through an orifice.


Vertical Axis
Positions of Anemometer in centre of concentric areas
Horizontal Axis
Approximate plane of meter readings
6"
8½"
10⅜"
12"

Elevation showing Method of Dividing Areas.

Section of Test Spigot

FIG. 19

To the right of the Fig. 17 is the layout showing the relative angles of blades and guide vanes. The actual angles are mentioned, but it is not known whether these apply to all such fans. The chief observation, however, is that whereas in theorizing for the curved blade propeller we had to assume that the air stream entered the wheel without shock, in this case such a condition is assured by the guide vanes.

Construction. The fan wheel is composed of a central casting having an overhanging peripheral rim of sufficient width to accommodate the blades. These latter are numerous and differ in number with the diameter of the fan, and it will be seen from the right-hand diagram that they overlap each other by about half their width. The rim of the wheel—and therefore the root of the blades—is slightly conical, whilst the tips also taper a little towards the discharge side, thus following the general lines of the other portions of the system.

Pressure Maintained. The fan appears to give very good results when working at a pressure of about 0·8 oz. per sq. in. (1·4 in. of water gauge), but pressures up to 1·2 oz. per sq. in. (2·07 in. of water gauge) are also recorded.

Uses. One of the chief purposes to which this fan has been put is for ship-board work, where its compactness is a distinct advantage. It can be introduced into a system of trunking without the enlarged feed box that would be necessary in the case of the ordinary propeller fan.

TESTING OF PROPELLER FANS

Unfortunately, no standard method of testing fans has yet been established in this country, although it is hoped that some recognized code will be inaugurated by the Fan Standardization Committee in the near future.

The commonest method employed for propeller fans is to measure the air velocity by an anemometer, and multiply the result by the area of the fan. Given proper conditions, this system is quite sound and fulfils all the requirements for non-pressure work.

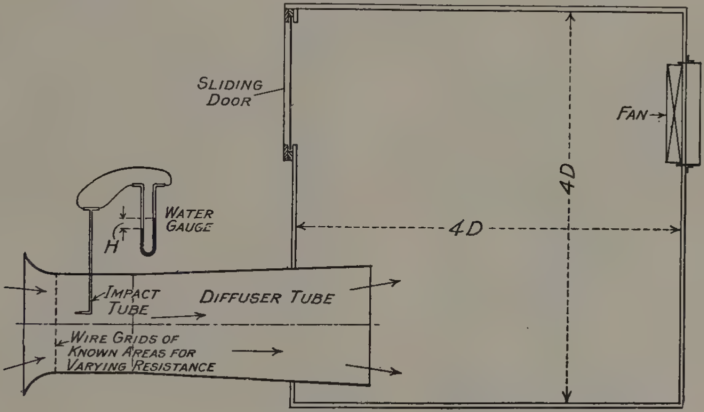


FIG. 20

A suitable air meter by Messrs. Negretti and Zambra is shown in Fig. 18. This instrument has the recording dials arranged in the centre of the vane system, and so offers no obstruction to the free movement of the air, which cannot be said of all such meters. The wheel is, in effect, a small fan having very light and well-balanced vanes, and the air rushing through the wheel causes it to revolve, the speed of rotation depending upon the linear velocity of the air current. In action the instrument is a fan reversed.

When the anemometer is purchased the makers should indicate a calibration factor by which all results have to be multiplied, and, if the instrument is used extensively, this calibration should be checked by them at suitable intervals.

The simplest way to take velocity readings is to fix a short circular spigot to the delivery side of the fan of the same diameter as the fan wheel, or as near to this diameter as possible. This spigot should be divided into a number of equal imaginary concentric areas, and four readings taken at each area, two on the vertical axis and two on the horizontal axis. The number of areas over which the test is to be made depends upon the size of fan and the degree of accuracy desired, for obviously, the larger the number of readings recorded the greater will the accuracy of the result be.

The diagram in Fig. 19 shows the suggested division of a 24 in. diameter tube with the positions of the anemometer indicated by dots.

The length of the spigot should be about equal to the diameter of the fan, and the measuring instrument must be placed actually inside the tube when readings are being taken.

The air volume being delivered will be represented by the mean of the velocities multiplied by the area of the spigot tube.

In order to take tests on the suction side of the fan, a large air-tight feed box is necessary, having an aperture at its back end across which the velocity readings are taken. This aperture should be fitted with a sliding door, so that any resistance can be put into the system by restricting the orifice. In this way, when readings are taken with a restricted orifice, the behaviour of the fan can be predetermined if it has to do duty under other than "free air" conditions.

The maximum area of the orifice should be greater than that of the fan, so that a perfectly unimpeded flow can be dealt with.

The "set up" is shown in Fig. 20, the box measuring four diameters of the fan in all directions. These dimensions may be exceeded.

however, and make no difference to the test results so long as the box is *airtight.*

Beneath the sliding door is indicated a diffuser forming the basis of another method of taking test readings of propeller fans. This is known as the *impact tube test,* and consists of a *pitot tube* (impact tube) which registers the dynamic head or velocity head of the air current by depressing a column of liquid in a glass U-gauge. From the difference in levels H of the two columns of liquid the air speed can be calculated.

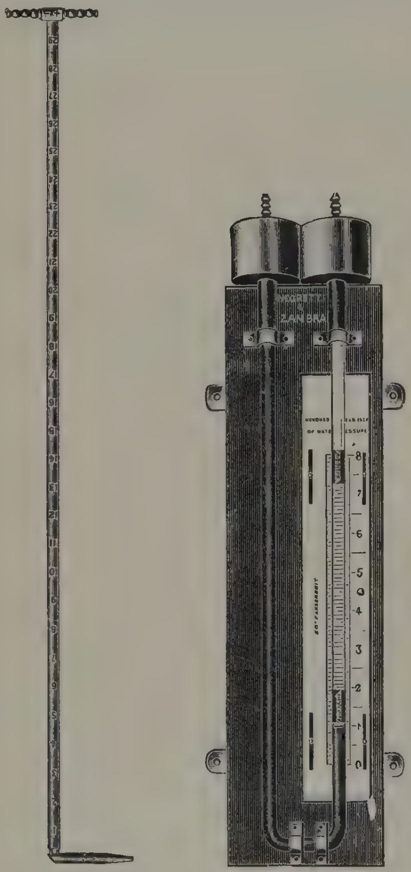


Fig. 21 Fig. 22

Pitot Tube and Draught Gauge

An illustration of a pitot tube and draught gauge, both by Negretti and Zambra, are shown in Figs. 21 and 22. The former is composed of a double tube, one within the other. The outer tube is graduated externally to facilitate the taking of readings at predetermined points across the air conduit. The small bore tube, which has its orifice at the nose of the instrument, registers the total impact pressure, whilst the outer tube registers the static pressure of the air, the difference between these two factors is velocity head and, as each tube is connected to one side of the draught gauge, the difference in levels of the liquid must indicate the intensity of the velocity head from which the air speed is calculated from the well-known formula for falling bodies, viz.—

$$v = \sqrt{2gh}$$

where v = velocity in feet per second
h = head in feet
g = acceleration due to gravity = $32{\cdot}2$ ft. per second.

In our case h is the equivalent head of the air column, due to its velocity. The head is, however, transmitted to the liquid within the draught gauge; so obviously, as this liquid is heavier than air its relative density must be considered, or h

$$= \frac{\text{density of liquid} \times H}{\text{density of air} \times 12}$$

where H is the difference in height of the liquid column in inches.

The most commonly used liquid for fan work is water, and assuming this to be at a temperature of 50° F., its weight per cubic foot will be $62{\cdot}409$ lb. therefore,

$$h = \frac{62{\cdot}409 \times H}{0{\cdot}07788 \times 12} = 66{\cdot}77\,H$$

Including this in our "falling bodies" formula we get

$$\begin{aligned} v &= \sqrt{2g \times 66{\cdot}77\,H} \\ &= \sqrt{64{\cdot}4 \times 66{\cdot}77\,H} \\ &= 65{\cdot}57\sqrt{H} \text{ ft. per second.} \end{aligned}$$

The tests above described are not a measure of air speed through the actual fan wheel and, when making calculations for H, the volumes obtained by the tests must be divided by the effective area of the fan wheel.

Readings with the pitot tube must be taken at several points on the vertical and horizontal axis, and as the area of the instrument at the point of impact is so much less, a greater number of readings than those suggested for the anemometer are advisable.

The purpose of the bell-mouthed and expanding diffuser is to prevent swirling and to assure an even flow of the air across the measuring instrument.

OUTLINES OF AUTOMOBILE ENGINEERING

By J. L. Milligan, B.Sc. (Vict.), M.I.A.E.

LESSON III

ENGINE DESIGN AND CONSTRUCTION

From the foregoing lesson it will be noted that if the bore and stroke of an engine are fixed the only factors which can increase its output are : indicated mean effective pressure, mechanical efficiency, revolution speed.

There are other qualities requisite in an engine which are of importance as well as power output for given size. Weight, cost, wear, fuel consumption, smooth running, flexibility: this latter is a much abused word. It is used here in the sense of giving an even turning moment or (b.m.e.p., which is proportional) over a wide range of engine speeds, in contrast to an engine which may not run well either below or above certain limits. In automobile engines weight is not an item of prime importance. It is of importance in so far as it increases production costs. Initial cost is of great importance, especially in the smaller vehicles. Wearing qualities are of first importance in public service and commercial vehicles, and in the larger sizes of private cars where the mileage is great, but, in general, smaller cars are not designed with this end in view.

Fuel consumption may be measured in miles per gallon or gallons per hour. The former is generally adopted. In commercial vehicles comparisons are made by using ton-miles per gallon, but as the wind resistance is about the same for all sizes of lorry, and all buses of same type have the same exposed front, comparisons are difficult unless speeds are low enough to avoid wind effects. If by any means fuel heat can be more cheaply converted into work on the piston, the increase of this work will mean no increase of fuel supply. The question of revolution speed is more difficult. In order to avoid vibration, six cylinders are necessary. Inertia and centrifugal forces rise as the square of the speed, and increased strength and stiffness are required. These increase cost, but enable better power output to be attained, and in the case of large engines the multiplication of small parts may reduce the cost as against a few large parts, as it may mean a more economic use of machinery and labour.

Flexibility. The larger the number of cylinders, the more flexible the engine is, as a general rule. The factors which make for high fuel efficiency and high output are often against flexibility, but this is not general nor desirable. In dealing with engine design we will consider those designs which aim at a high power output for given size, with good consumption and medium cost, indicating where costs can be saved at the expense of performance.

VALVES

It is obvious that the entry and release of the gases by the valves is a vital point of engine design. Valves of two types are in use. The commoner is called a mushroom or poppet valve, shown on Fig. 13. It must be of a material which withstands high temperature without burning or corrosion, and must be free of distortion and have high strength (both tensile and compressive) at these temperatures. The valve is closed by a spiral spring and its operation must be positive. The spring must be strong enough to overcome the inertia of the valve at the top of its lift, and also strong enough to overcome this inertia and the friction in the guide during its fall.

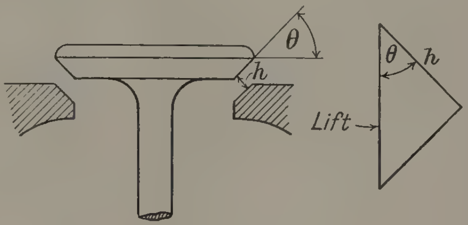


Fig. 13

The lift, full opening and closing occupy at 3,000 r.p.m. only ·013 sec. In order to pass gas effectively at high speeds the valve must be large and have a high lift. This all throws more work into the spring. All sorts of springs have been used, but the spiral spring in compression has now a monopoly.

It will be seen that these improvements of the valve all lengthen the distance from the valve face to the end of the stem. The other type of valve is the sleeve valve, described later.

Valves have been fitted in many ways, but they can be divided into two types, side valves and overhead valves (Figs. 14 and 15), also hybrids having one side and one overhead. Modern side valves are both on one side of the cylinder and are operated from a camshaft carried in the crank casing, all the valves of the various cylinders being operated from one shaft.

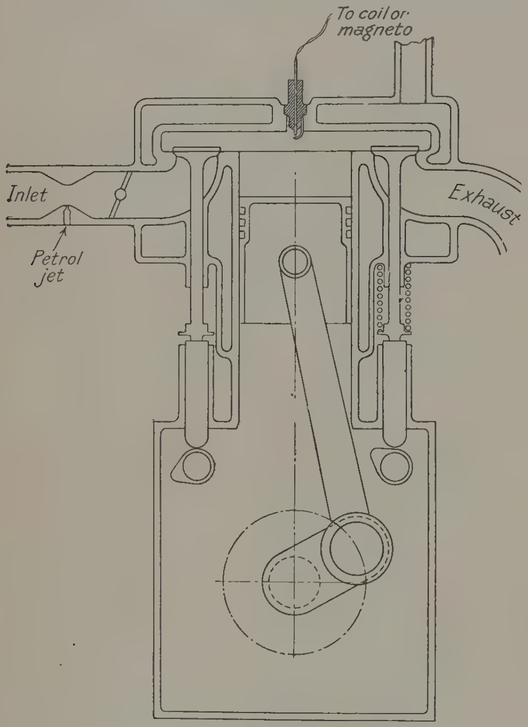


Fig. 14. Petrol Engine with Side Valves

Note. In this diagrammatic sketch, pieces which are made in more than one part for manufacturing reasons are shown in single parts.

The cams usually act on sliding tappets, which are adjustable for length and sometimes have a roller to run on the cam. Some makers use hinged arms called rockers, between cam and tappet to take the side thrust of the cam. In order to clear the connecting rod end and yet keep the valves close to the cylinder bore the valves are often inclined. Overhead valves are operated either by an overhead camshaft or two overhead camshafts (one for inlets, one for exhausts), or by a camshaft in the crank case and a system of rods and levers (Fig. 16). The former is heavy, as no part of the crank case can be used to carry the camshaft, and its height above the engine makes the design look clumsy. The lubrication is usually either too little or too much, and connections for the cylinder water jacket are not easy to make. The push rod system is heavy, as the rod and lever work has to have springs fitted to make it follow the cam and relieve the valve spring of all work, except moving the valve itself. It does not present much trouble with camshaft drives, and is quite a popular system.

The one up, one down, type ~~does not~~ present any advantage to designers—it has to have push rods for the overhead valves.

Valves must be accessible to inspect and to true up the working face of valve and seat. When overhead valves are used the cylinder head is detachable, and the valves and their seats are removable after the head is removed.

When side valves are fitted the head is often removable and the valves are then accessible, or separate screwed caps are fitted above each valve. The removal of the cap, which must be larger than the valve, allows of the removal of the valve. Of recent years the detachable head, once common, and then abandoned by European designers, has returned to favour, chiefly because of better machining facilities, and the possibility of machining the whole combustion chamber with accuracy, so obtaining uniform compression in all cylinders. Overhead valves allow of the combustion chamber design being carried out almost unhampered. That is to say, the combustion chamber can be designed to present the smallest possible surface for a given volume. With side valves there must be plenty of room for gases round the valves and over them and, with large valves, this gives a flat shallow space not conducive to good combustion.

If an overhead valve breaks, the piston is usually broken: this was the standing objection for many years to overhead valves. The side valves of that time being small, means were often provided to prevent the entry of a valve head into the cylinder itself. In modern engines there is no room for such fixtures and the risk, a very small one, has to be taken. There is, therefore, no objection to overhead valves on this ground.

The principal objection to overhead valves lies in the noise caused by the striking of tappets on valves and of valves on their seats. These noises are hard to deal with because the mass of metal present is small, whereas in side valves the tappet noises are in proximity to a heavy casing and are damped out. The driving mechanism also requires careful design if noise is to be avoided.

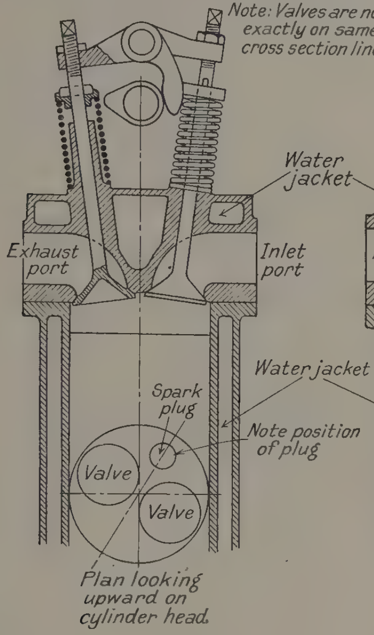

Note: Valves are not exactly on same cross section line.
Water jacket
Exhaust port
Inlet port
Water jacket
Spark plug
Note position of plug
Valve
Valve
Plan looking upward on cylinder head.

FIG. 15. OVERHEAD VALVE WITH OVERHEAD CAMSHAFT

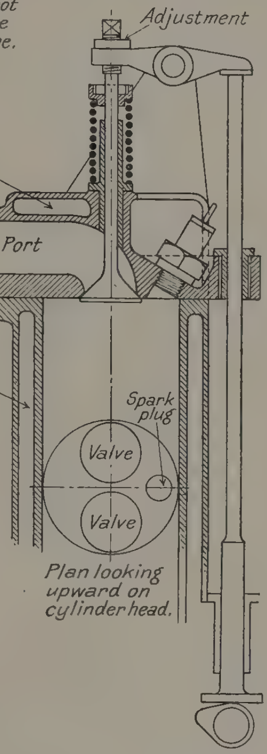


FIG. 16. OVERHEAD VALVE WITH PUSH RODS

SIZE OF VALVES

If a valve, Fig. 13, of diameter d has an opening h = lift × cos θ. The area through the valve when open $= \pi \times d \times h$.

The area of the pipe is $\frac{\pi d^2}{4}$.

So that to obtain the same area through the valve as through the pipe, $h = \cdot 25d$. If the angle of the valve face is 45°, then lift $= 1{\cdot}41 \times$ opening, so lift $= \cdot 35 \times$ diam. of valve.

This is the usual formula for valves on steam pipes, etc., but is not followed for automobile engines. The lift is usually ·25 to ·3 of the diameter.

It will be observed on turning an engine round that the piston travels slowly at the ends of the stroke and quickly in the middle. If gas is to follow up the piston on the inlet stroke, then the ratio of gas speed to piston speed will be the inverse ratio of valve area to piston area. It is not usual to consider maximum gas speeds, but average gas speeds, that is, to assume that the cylinder intakes the swept volume during one stroke, and work out the mean gas speed which would result from this. These figures are merely for comparison of engine with engine: they are not measurable.

Consider an engine 100 mm. bore × 150 mm. stroke, swept volume 1,078 cub. cm. or 66 cub. in. running at 1,200 rev. per min., and having a valve $1\frac{1}{2}$ in. diameter with $\frac{1}{2}$ in. lift bevel at 45°.

The area through the valve opening is 1·68 sq. in. The amount of gas passed in one induction stroke is 66 cub. in. above. The duration of one induction stroke is half a revolution or $\frac{1}{40}$ sec. The problem then becomes 66 cub. in. of gas passes through an opening 1·68 sq. in. section in $\frac{1}{40}$ sec., what is the mean velocity in feet per second?

$$= \frac{66 \times 40}{1{\cdot}68} = 1{,}560 \text{ in. per sec.}$$
$$= 130 \text{ ft. per sec.}$$

which is a reasonable velocity.

The exhaust valve is generally the same size as the inlet. There have been eminent designers who made exhaust valves larger than inlet valves, and some equally eminent who maintained the opposite view. As equal valves make manufacture simpler, they are now regular practice. It is found that several small valves are lighter and give better results than one large one, and hence on some racing engines the valves are duplicated.

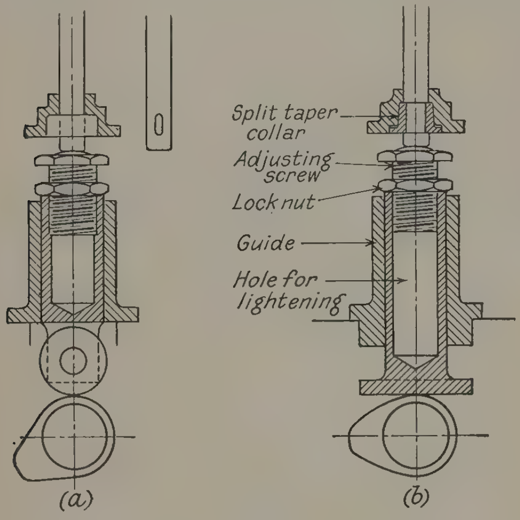


FIG. 17

The results are not so much better as to justify the extra cost except for special purposes.

Valves have been made of many shapes. A few are shown. Fig. 14 is common. Many materials have been used. Chrome steels, owing to their freedom from corrosion, are at present in most demand, especially cobalt-chrome steels, which are difficult to forge and turn, but give long life under severe conditions. The neck of a valve requires to have a large radius to prevent the head breaking off. It is sometimes recessed inside as Fig. 15 (the tulip valve) to lighten it. Valves should have a slot across the head to take a screwdriver for grinding in purposes.

Many arrangements have been in use to hold the spring at the foot. The plain cotter (Fig. 17 (*a*)) is very largely used and quite sound. Fig. 17 (*b*) shows another type. Both have the same difficulty, they weaken the valve stem.

Tappets are of many sorts. Rollers give good results and long life, but are often omitted for cheapness and quietness.

The design of cams is too involved and contentious a subject to be dealt with here at length. Many designers try to obtain a very rapid lift and closing for the valves. This is not so easy to obtain with a roller as with a flat tappet, a flat tappet being the equivalent of a roller of infinite diameter. The objections to rapid lift and drop are the excessive spring pressures necessary to prevent the valve continuing to lift, the noise, and the uncertainty of closing. The objection to slow lift and drop are the restrictions on gas flow.

It should be noted that the flat or tangential cam is used with roller tappets and the rounded cam with flat tappets (Fig. 17).

Valve springs, as stated, are of round wire in spiral form, and are usually of carbon steel, specially treated.